ELECTROMECHANICAL FLYWHEEL BATTERY
EDU DEVELOPMENT PROJECT

Grant No. MDA972-93-1-0025

FINAL REPORT

DECEMBER 1996

Prepared By:

American Flywheel Systems, Inc.
P.O. Box 449
Medina, WA 98039

The project reported herein was sponsored by the Defense Advanced Research Project Agency. The content of the information does not necessarily reflect the position or the policy of the Government, and no official endorsement should be inferred.

ELECTROMECHANICAL FLYWHEEL BATTERY
EDU DEVELOPMENT PROJECT

Grant No. MDA972-93-1-0025

FINAL REPORT

DECEMBER 1996

Prepared By:

American Flywheel Systems, Inc.
P.O. Box 449
Medina, WA 98039

The project reported herein was sponsored by the Defense Advanced Research Project Agency. The content of the information does not necessarily reflect the position or the policy of the Government, and no official endorsement should be inferred.

American Flywheel Systems EDU Program - Final Report

Table of Contents

List of Figures

List of Tables

1.0 Introduction to AFS' SMUD/ARPA Program

In January 1993, American Flywheel Systems, Inc. (AFS) initiated a program to develop and commercialize its proprietary Electro-Mechanical Flywheel Battery (EMFB) for commercial deployment in electric vehicles. In May 1993, AFS executed a contract with Honeywell's Satellite Systems Operation (HSSO), AFS' primary engineering and technology development contractor, to design, build and test a prototype EMFB. As an intermediate step, a pre-prototype EMFB (or Engineering Development Unit [EDU]) of the EMFB would be constructed. The EMFB EDU is a critical step in the AFS EMFB development and commercialization process. The development and testing of the EDU would clearly affirm and demonstrate the technical viability of the EMFB, while at the same time, provide information essential to the development of a full-scale EMFB prototype. The EDU program was designed to provide sufficient and necessary structural and operating data on the EMFB components and system to ensure that the subsequent EMFB prototype would be technically suitable for commercial-scale testing. At program commencement, the total cost to develop and test the EDU was estimated to be less than $5 million, with EDU completion expected in mid 1994.

Building a viable, reliable, safe and cost effective EMFB is challenging from any perspective. All of these challenges/issues must be simultaneously addressed and successfully overcome for an EMFB to be practically deployed in electric vehicles. However, by 1993 contemporary flywheel feasibility studies (e.g., Reference 2-1) had concluded that the technology was available to develop an EMFB, and that the challenges were those of additional adaptation and system integration.

The cornerstone of the EMFB program is based on recent technology innovations in the areas of materials and electronics, including:

- **Materials.** Advanced high strength composite fibers for flywheel rotor and containment vessel, and high power density permanent magnets for bearings the and motor/generator.

- **Electronics.** Very large scale (VLSI) digital controllers for active bearings and the motor/generator.

- **Bearings.** High-performance, low loss bearings, including passive and active magnetic bearings and ceramic bearings.

In addition to HSSO, AFS engaged several individuals and organizations to assist in the AFS EDU Program. Through a "Work for Others" contract, AFS engaged Oak Ridge National Laboratory (ORNL) of the U.S. Department of Energy in April 1994 to design, fabricate and test flywheel rotor subsystems for the AFS EDU. In addition, Professor Paul Allaire, a widely acknowledged expert in the field of magnetic levitation

and leader of the magnetic suspension program at the University of Virginia, was contracted to assist AFS in the development of the magnetic bearing for the EDU.

AFS was awarded a patent for the EMFB system (1993). AFS designed its EDU program approach to use its patent as a foundation to develop new technology for its EMFB. AFS was committed to finding the best talent and resources to ensure that its commercial EMFB is cost-effective, reproducible, reliable and safe for use in personal electric vehicles. The EDU program approach was comprised of the following technology developers:

- HSSO, the primary development contractor, was responsible for the design, fabrication, and testing of the EDU and all components and systems contained therein. HSSO, located in Phoenix Arizona, was the principal investigator and technology developer for AFS. In October 1994, HSSO ceased all work on the AFS EDU, and effectively terminated its responsibilities under the AFS EDU program.

- Martin Marietta Energy Systems, Inc. (later subsumed into Lockheed Martin), the contracted operator of the DOE's ORNL, was contracted by AFS under the U.S. Department of Energy's Work For Others program. With agreement between all parties, ORNL under HSSO's technical direction would design, fabricate, and test the EDU rotor system. Following HSSO's unexpected departure from the AFS EDU program in October 1994, AFS authorized ORNL to take on many of the EDU system design and testing responsibilities previously required of HSSO.

- In March 1995, AFS engaged Dr. Paul Allaire, Professor of Engineering, University of Virginia, and independent consultant, to design the AFS EDU magnetic bearing suspension system. While AFS completed its review of the magnetic bearing design developed by HSSO, Dr. Allaire (with other highly skilled and experienced consulting engineers at the University of Virginia) developed an alternative design to that of HSSO. Each design commissioned by AFS has unique features and attributes suitable for use in EMFBs ranging from the fundamental design philosophy to the levitation controls specified for idle EMFB operation.

AFS not only engaged highly qualified technology and system development contractors to support the development of the AFS EDU, but it assembled well-qualified and widely experienced consultants to lead and monitor EDU development. In addition to Mr. Edward W. Furia, AFS' Director of Programs, AFS engaged Dr. Edward S. Zorzi, former General Manager and Director of Mechanical Technologies, Inc., to be the AFS Contract Manager, and Dr. James H. Williams, President and CEO of RDC, Inc., to be the AFS Contract Administrator. The engineering and management skills of several talented AFS consultants including Mr. John V. Coyner formerly with Martin Marietta (ORNL), Dr. Robert B. Bartlett formerly with Fairchild Space and Defense Corporation, and Mr. Ray Griswold formerly with HSSO were profoundly important to the technical progress made during the AFS EDU program.

Despite the contract and institutional problems that occurred during EDU development heretofore, AFS has made enormous progress toward a commercially viable EMFB. The following sections describe the EDU program's approach, technical activities, and accomplishments to date.

2.0 Executive Summary and Key Accomplishments

2.1 Introduction

In January 1993, American Flywheel Systems Inc. (AFS) initiated a program to develop and commercialize its proprietary Electro-Mechanical Flywheel Battery (EMFB) for commercial deployment in electric vehicles. The operational prototype was to demonstrate the technical viability and operation of the Electro Mechanical Flywheel Battery (EMFB) as a commercially feasible replacement to conventional chemical batteries. As an intermediate technical step in that process, and to provide an opportunity to evaluate a pre-prototype EMFB, an Engineering Development Unit (EDU) was also to be constructed.

EMFBs present the prospect of providing an environmentally compatible energy source offering outstanding life and performance for electric vehicles. Contrasted with the chemical species of battery, the EMFB is not an electro-chemical device. Instead it is an electro-mechanical system that efficiently converts (through its motor-generator) the mechanical kinetic energy stored in a high speed rotating flywheel rotor into usable electrical power. Recharging the EMFB reverses that energy exchange and the battery stores supplied electrical energy as mechanical kinetic energy as the motor spins-up the flywheel rotor to speed. Operating in a vacuum to minimize losses, and enclosed in a safety vessel, the EMFB is a mechanical battery, not a chemical battery, and therefore is not sensitive to the typical thermal degradation and other life limiting factors inherent to chemical batteries. A successfully developed EMFB has the potential of offering high specific energy (multiples above lead acid batteries) and specific power (sufficient to provide outstanding acceleration and capture the available energy of regenerative braking) in an environmentally benign package.

The cornerstone of the EMFB program is based upon recent technology innovations in the areas of materials and electronics, which include:

- Materials: Advanced high strength composite fibers for the flywheel rotor and containment vessel, and high power density permanent (rare earth) magnets for the bearings and motor / generator.
- Electronics Very large scale (VLSI) digital controllers for the active bearings and motor / generator.
- Bearings high-performance, low loss bearings, including passive and active magnetic bearings and ceramic bearings.

The challenge of building a viable, safe and cost effective substitute to a chemical battery using an electro-mechanical device that rotates at hundreds of thousands of revolutions per minute is certainly a significant undertaking from any prospective. Add to that the complexity of having the mechanical battery tolerate abusive road shock and vibration characteristics typical of modern automobiles, and one begins to comprehend the overall challenge of developing the EMFB. The AFS patents, first granted in June 1992, offered the technical foundation for an electro-mechanical flywheel development and refinement program. Those documents addressed the principle components and design of the EMFB including the; composite rotor, motor/generator, magnetic and backup suspension systems, vacuum/safety enclosure and electronics/controls. The AFS patent also addressed specifics in dealing with gyroscopic effects resulting from a spinning mass for mobile deployment in an electric vehicle. It offered a viable technical genesis and/or point of departure for this undertaking.

2.2 Executive Summary of Program Activities

In early 1993 contemporary flywheel feasibility studies, (e.g. Reference 2-1)[1] had concluded that the technology was in place to develop such a device (i.e. the EMFB) and all that was required was additional engineering and careful systems integration. As all of the components of the EMFB had been used in other applications in some form, this is precisely the approach taken by AFS' prime contractor, Honeywell Satellite Systems Operation (HSSO), when the development program was proposed in December 1992. The program provided for three critical and timely hardware deliverables. These were:

1. A Demonstration System suitable for illustrating the basic principals of the EMFB,
2. An Engineering Development Unit, (EDU) which offered an expanded test platform for the first generation system integration, and
3. A fully functional EMFB Prototype for an electric vehicle.

[1] References per Section 3.2

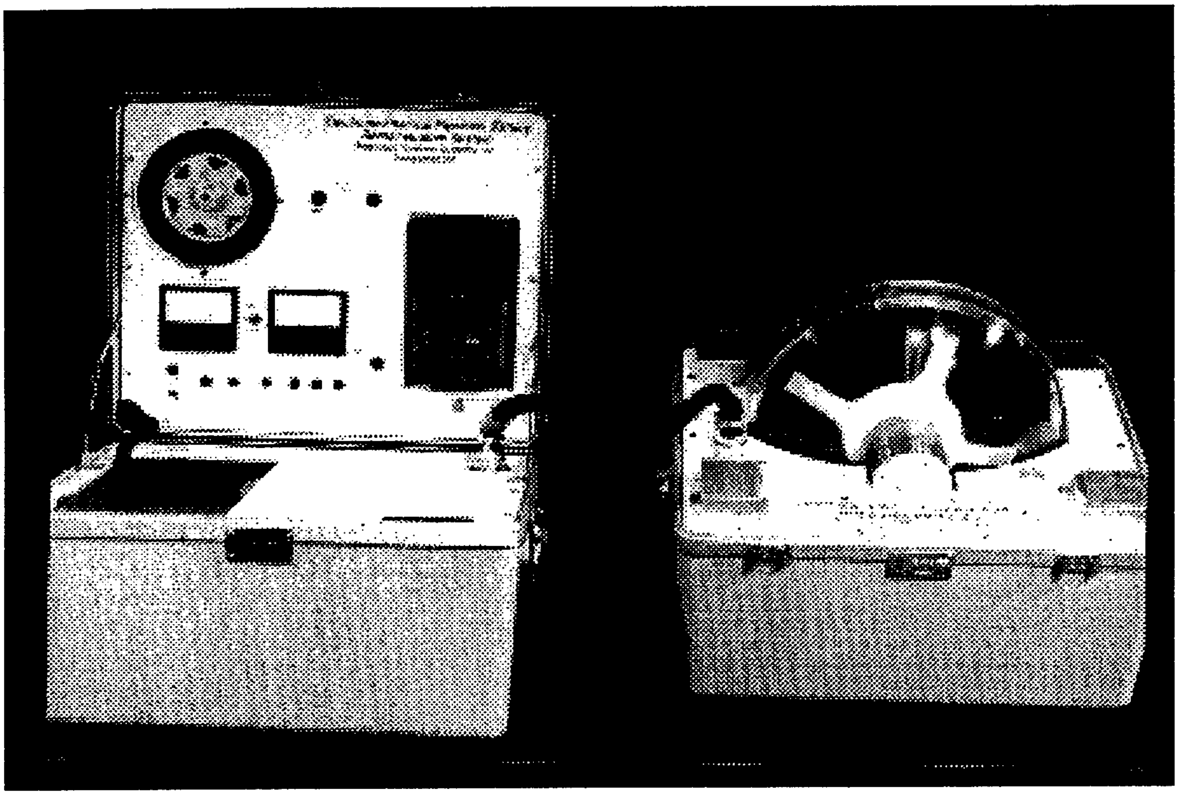


Figure 2-1 AFS Demonstration System

The HSSO proposal stipulated that the demonstration system was to be delivered within three (3) months of program start and the EDU was to be delivered twelve to fourteen (12 to 14) months after commencement of this effort. The prototype would be delivered in approximately one year following EDU delivery. As the program launched, and in particular as this technical undertaking was better comprehended, HSSO advised AFS of the desire to re-plan the proposed effort. Honeywell subsequently offered a re-plan submittal to AFS in November 1993, which was thereafter submitted for approval to SMUD by AFS in March 1994. The re-plan maintained all of the key EDU requirements, save the final design of the safety - containment vessel which would be applied only to the prototype EMFB, and offered EDU delivery on an eighteen (18) month schedule. This permitted HSSO to take full advantage of the flywheel rotor burst testing scheduled to be completed at HSSO's sub-contractor, Oak Ridge National Laboratory (ORNL). Under the Honeywell effort the EDU test fixture was completely designed and fabricated for component integration and testing. An EDU test plan and procedure were also developed including full instrumentation. The motor/generator was tested to 63,000 rpm. Shortly after this test, HSSO ceased their technical efforts on the AFS program.

After it was determined that HSSO would not be expected to resume the technical efforts in a timely fashion, the EDU was re-defined to be a fully instrumented test rig capable of running the rotor to full speed on an EMFB bearing suspension system at the

ORNL spin-pit facilities. ORNL took on the added commitment to provide not only the rotor but also the fully instrumented test rig integrated with their air drive turbine equipment and capable of being operated in their spin pit test facility. The bearings which were under development at the University of Virginia (UVA) for AFS were candidate suspension components to be integrated into the test rig at ORNL, however the backup ceramic bearings were to be applied for the earlier series of EDU tests at ORNL. The EDU definition has therefore evolved as the AFS program was refined.

2.3 Accomplishments

Table 2-1 provides a summary chronology of pivotal program events and accomplishments. Some of the details are AFS proprietary information and cannot be disclosed herein. However, the authors have made every effort to offer complete disclosure within the bounds of those constraints.

Table 2-1 Summary of Pivotal Program Events

Date	**EDU Accomplishment**
Year 1993	
August	Demonstration system delivered to American Flywheel Systems, Figure 2-1.
September	System engineering plan completed.
November	Honeywell re-plan submitted to AFS for approval. Restructure EDU for progressive component "build-up" and testing to completion.

Year 1994	
January	Optimization analysis and software operational - numerous EMFB / EDU designs evaluated ranging from 2 kWhr to 14 kWhr for electric vehicle deployment. HSSO EDU test fixture designed per Figure 2-2.
February	Five (5) patent disclosures transmitted to AFS, including; 1. Composite flywheel rim and hub interface 2. Gimbal mount for flywheel 3. "Ganged" gimbal system for flywheels 4. Burst containment vessel 5. Energy absorbing housing.

March	One (1) patent disclosure transmitted to AFS on the magnetic bearings.
April	Re-plan transmitted to SMUD for approval by AFS. Request to go forward with the Honeywell EDU activity.
April	One (1) patent disclosure transmitted to AFS on the motor/generator.
April	3.6 kWhr / 8 kW EMFB design selected by AFS.
June	EDU test plan developed.
June	Four (4) patent disclosures transmitted to AFS, including; 1. Power converter 2. Magnetic bearing 3. Motor/generator 4. Vacuum seal.
July	EDU structural dynamics report transmitted to AFS.
August	Outgassing materials test report transmitted to AFS.
August	Draft patent disclosure transmitted to AFS on EMFB system.
August	Tribology test report (pin-on-disk) transmitted to AFS. Touchdown bearing mechanism and materials selected.
September	EDU design review with Oak Ridge National Laboratory personnel.
October	Motor / generator operated to 63,000 rpm in EDU test facility
December	Customized high speed ($100,000^{+}$ rpm) drive turbine procured for spin testing

Year 1995	
February	One (1) patent disclosure transmitted to AFS by ORNL for a composite rotor.
April	AFS transmitted a final report to SMUD on magnetic bearings.
April	AFS transmitted a final report to SMUD on motor / generator.
April	Selection of EDU bearings for rotor testing.
April	Modified EDU established - technical review meeting at Oak Ridge National Laboratory.
June	Assembly tooling for composite rotor completed
August	Composite rotor - display unit delivered to AFS
September	Assembly of composite rotor
September	University of Virginia study and report on self-scheduled controllers for EMFB magnetic bearings.

Year 1996	
March	Completed design of magnetic bearing test rig.
April	Completed all mandrels and tooling for rotor fabrication.
May	Rotor dynamics evaluation of test rig completed by ORNL.
September	Final report on magnetic bearing controllers.

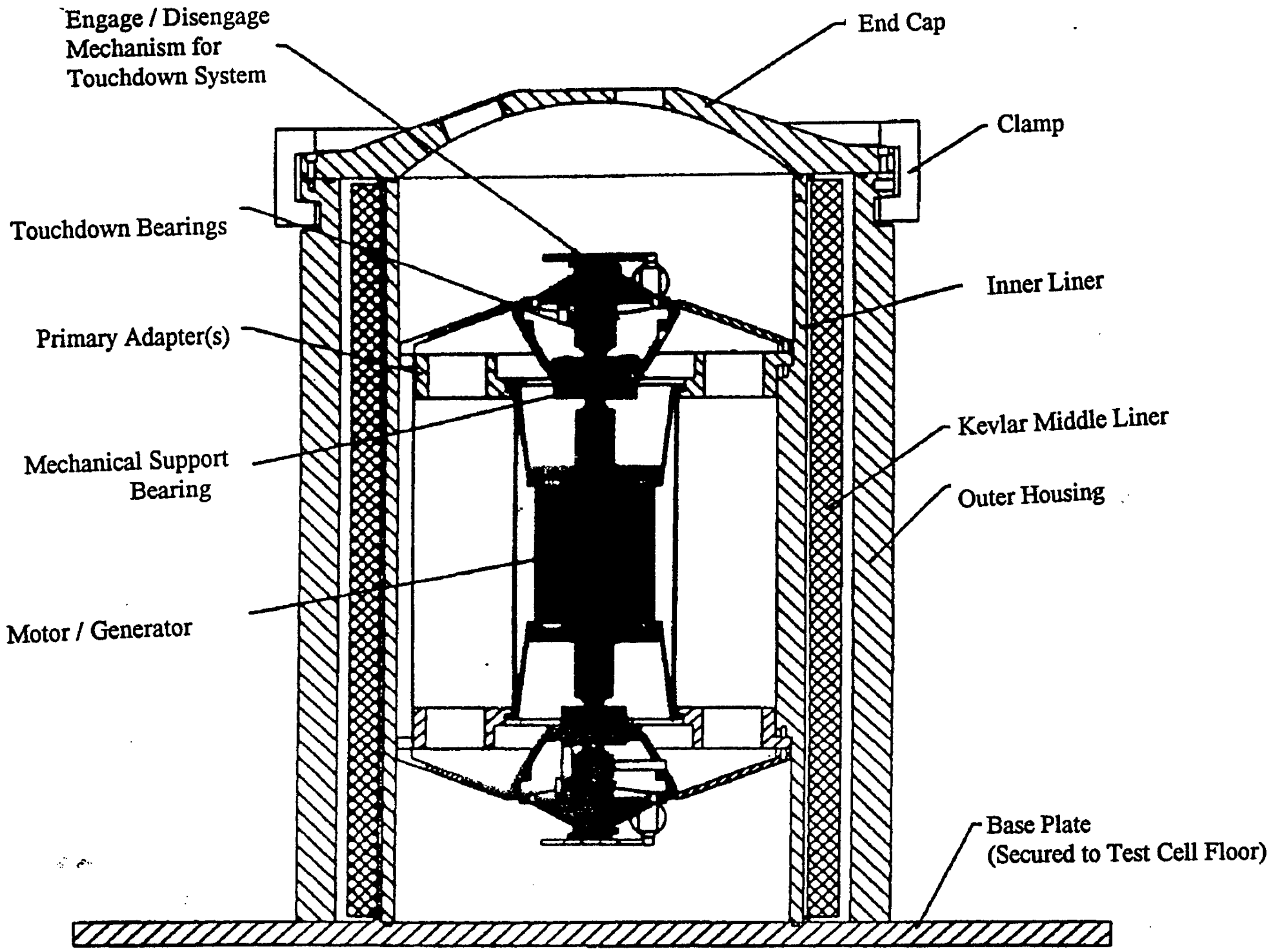


Figure 2-2 HSSO Design EDU Test Fixture

The EDU test assembly, as finally configured, consists of the rotor, bearings and appropriate structure to be driven by a 100,000^{+} rpm Barbour Stockwell air turbine, suspended within the ORNL spin pit at Oak Ridge TN. The EDU is fully instrumented with pressure, proximity, thermal and vibration sensors etc. and capable of monitoring and documenting the performance of the composite rotor. A test sequence includes shakedown, full speed operation, and overspeed testing as well as intentional burst testing of the composite rotor. See section 3.1.5. of this report for further details of the Oak Ridge National Laboratory program effort.

3.0 Program Approach

3.1 Introduction and Background to AFS efforts

AFS Patent #5124605 dated June 23, 1992, was developed from extensive analytical and experimental evaluations, as detailed in the AFS report, Reference 3-1. That report describes the EMFB and suggests technical approaches for its development. The flywheel system as characterized therein was intended for application in a mobile environment, as close attention to issues such as gyroscopic effects, etc., were addressed in that work body of knowledge. The AFS patent focused upon the concept of producing a very high specific energy (Whr/lb) EMFB and identified the major components of the system which are, per Figure 3-1:

1. **The flywheel rotor** which is manufactured of high strength fibrous composite materials and which spins at high speeds to store the kinetic energy (energy stored is a function of the square of the rotor's speed).
2. **The high efficiency motor-generator** which converts rotational kinetic energy from the flywheel to electric power and also serves to recharge the flywheel rotor.
3. **The low loss bearings,** generally non-contacting magnetic bearings which minimize the mechanical losses of the rotating system.
4. An addition bearing set that is termed the **"touchdown or backup bearings"** offering protection to the system under high or abusive shock and vibration conditions.
5. **The controls and electronics** for component operation, system health monitoring, power delivery and system interface.
6. **A vacuum / safety containment chamber** which minimizes windage frictional losses (air turbulence) and also provides for a safety enclosure.

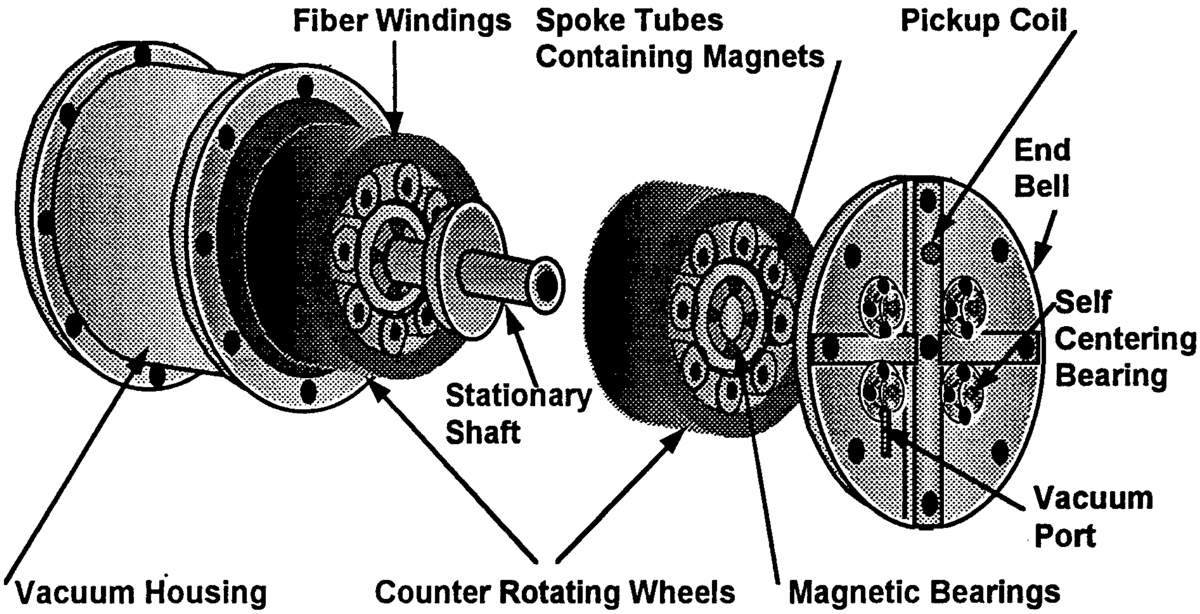


Figure 3-1 Electro Mechanical Flywheel Battery

The AFS Electro Mechanical Flywheel Battery is designed to replace chemical batteries and provide superior performance in electric cars, as well as in spacecraft and in various stationary applications. The mobility and efficiency required by electric vehicles implies that the system must operate very efficiently yet retain its inherent advantages of long life and immunity to degradation from thermal effects or numerous deep discharge cycles. The required efficiency is achieved by operating the system in a hard vacuum, suspending the rotating parts with magnetic bearings, and minimizing all electro-magnetic losses. High specific energy can only be accomplished by operating at very high RPM, as the stored energy is directly proportional to the inertia of the flywheel and to the square of the rpm. The rotor is made from the latest graphite epoxy materials and is a unique design that maximizes energy density while retaining dynamic stability at very high rotational speeds. By operating at high speeds excellent specific energy can be achieved, but it is noteworthy that this exacerbates losses from drag forces and requires special attention to safety issues. Eddy currents and induced magnetic drag have been nearly eliminated by unique AFS motor/generator and magnetic bearing designs which also greatly reduce another potential problem, heating. Since the system must operate in a hard vacuum it is difficult to remove the heat from the system. The high operating efficiency reduces this problem so that all cooling can be accomplished passively.

3.1.1. Program Challenges - Overview

Challenges are not in short supply when one is involved with the development of the first EMFB designed expressly for an electric vehicle. From a broader perspective these challenges encompass:

- **Safe operation** of the high speed flywheel including electric vehicle application of the EMFB.
- **Technically competitive** with current technologies (e.g. improved energy storage, power delivery, quiet, long term storage, etc.)
- **Cost effective** when compared to competitive technologies

These were the primary or institutional goals as established by AFS throughout the development activity. Although these goals are formidable, so is the potential for flywheel batteries to overcome the limitations of conventional chemical batteries. Certainly the advantages and opportunity for substantial performance improvements are apparent as one reflects upon the following technical advantages of the EMFB;

- **Compact** - efficient geometry
- **Long Life** - without replacement or major overhaul
- **Deep cycling** - deep discharge and rapid recharge
- **Power delivery** - high specific power
- **Energy delivery** - high specific energy
- **Environmentally benign** - no toxic effluents or disposal problems
- **Efficient operation** - and over a wide temperature range

Section 3.1.4.2. below provides a more detailed discussion of these challenges as they relate to the EMFB.

Therefore, the technology; (a) must be safe, (b) must perform and (c) must be cost effective if the EMFB is to become a commercially viable competitor to the chemical species of battery. It was understood by all involved in the AFS program that safety was the top priority which could/would not be compromised. This is why AFS requested HSSO to perform a full scale burst test of the composite rotor.

From a programmatic or management viewpoint, key challenges also included the direct coordination of multiple sub-contractors. At the commencement of this effort, AFS began the assembly of a unique technical and programmatic leadership staff to support and lead this project. Dr. Zorzi, former General Manager of the Engineering and Technology Division at Mechanical Technology Inc., joined AFS in April of 1993. Dr. Zorzi and Dr. Williams of RDC Inc. provided technical and programmatic leadership to AFS throughout the early stages of the HSSO contribution. Subsequent to Mr. Coyner's retirement from Martin Marietta, he joined AFS and assisted in the leadership of the composite development efforts at ORNL. AFS was also strengthened by the retirement of Mr. Griswold from Honeywell, as he was the HSSO Technical Director of the AFS program and continued his technical leadership, and involvement with AFS. Dr. Bartlett, formerly with Fairchild, also joined AFS to completed the AFS technical/leadership team on this program. During the execution of this program the principle sub-contractors included:

Table 3-1 Team Members

Team Member	Responsibility	Leadership
American Flywheel Systems	Configuration Management, EV Integration and Manufacture	Dr. Edward S. Zorzi / Dr. James Williams
Honeywell Satellite Systems Operation	System and Key Component Development	Mr. James Kiedrowski / Mr. Ray Griswold
Oak Ridge National Lab.	Composite Material Component Development	Mr. John Shaffer / Mr. Vince Campbell
University of Virginia	Magnetic and Back-up Bearings Development	Dr. Paul Allaire

3.1.2 Contractual Issues and Costs

After receiving a U.S. patent for its EMFB system in June 1992, AFS began the process of assembling the resources required to develop and commercialize an EMFB suitable for use in electric vehicles. AFS determined that the most productive approach in achieving its commercialization objectives was to acquire the best talent and resources available to manage and develop its EMFB. As with any major technology development activity, the acquisition and protection of intellectual property produced on behalf of AFS during the planned four year EMFB development program was a essential element in all contractual relationships. In addition, AFS maintained the contract flexibility to acquire new talent and resources through either primary or subordinate contracts as necessary.

3.1.2.1 Honeywell's Satellite Systems Operation (HSSO)

Prior to its Participation Agreement with SMUD in July 1993, AFS had awarded a contract to HSSO in January 1993 for nearly $5.5 million to be the primary developer of the AFS EMFB. Under its contract with AFS, HSSO would be responsible for development of an EMFB suitable for use in electric vehicles with an energy density of no less than 45 Whr/lb. The three primary HSSO contract deliverables during the 26 month program schedule were:

- A Demonstration System suitable for illustrating the basic principles of the EMFB
- An Engineering Development Unit (EDU) which offered an expanded test platform for the first generation system integration, and
- A fully functional EMFB Prototype for an electric vehicle.

With the initiation of the program, HSSO's obligations included the definition of a sub-contractor plan as well as an evaluation of the available technical resources which could apply to the development of the EMFB and EDU. To address this requirement, HSSO performed a state-of-the-art review and technical "due diligence" of available

resources. As a result of that effort, HSSO decided to rely upon their internal technical capabilities, along with those of the HSSO Durham, North Carolina facility, for the majority of component and system development. The major exceptions would be the composite rotor fabrication and the manufacture of the touchdown bearings, which were to be sub-contracted.

Before it ceased support of the AFS EMFB program in late October 1994, HSSO had delivered the Demonstration System, and made progress in the development of the EDU including fabrication and initial testing of a reduced-scale high-speed motor/generator, design of a rotor touchdown bearing system, and preliminary designs for the EDU, the magnetic bearing subsystem, and EDU test rig and containment. ORNL was selected as the provider of the flywheel rotors.

Based on a modified work plan submitted by HSSO in November 1993, AFS subsequently provided SMUD recommended EDU program changes along with an assessment of cost and schedule implications. In late 1993, HSSO had projected the first quarter of 1995 for its completion of the EDU fabrication and testing.

3.1.2.2 Oak Ridge National Laboratory (ORNL)

Based on the recommendation of HSSO and concurrence with SMUD, AFS initiated negotiations in October 1993 for a flywheel rotor fabrication contract with ORNL. The contract between Martin Marietta Energy Systems (later subsumed into Lockheed Martin) and AFS is a Work for Others (WFO) contract approved by the U.S. Department of Energy in late April 1994. With HSSO concurrence, the contract required ORNL to receive its technical direction from HSSO. Per agreement, ORNL would fabricate and test flywheel rotors per specification for the EDU within 12 months for an estimated $1.0 million.

After HSSO's departure from the AFS EMFB program in October 1994, ORNL assumed additional responsibilities. In addition to conducting material tests and fabricating and conducting flywheel rotor tests, ORNL agreed to integrate and test the complete EDU system. To date, ORNL has completed design and fabrication of the EDU rotor system components, and initiated assembly. To assist ORNL in the fabrication of the rotor system, AFS in the first quarter of 1996, procured mandrels required for winding composite fibers. In addition, AFS procured an air turbine from Barbour Stockwell designed specifically to operate and test the EDU at speeds exceeding 100,000 rpm. With SMUD's concurrence, AFS procured the necessary high-tensile strength composite fibers for ORNL's fabrication of the EDU flywheel rotor components.

3.1.2.3 University of Virginia (UVa)

With the magnetic bearing system being the primary determinant of EDU energy efficiency and a major contributor to EDU energy density (i.e., Whr/lb), AFS determined that an alternative magnetic bearing system design was essential to ensure EMFB objectives were achieved. With SMUD's concurrence, in March 1995 AFS contracted with Dr. Paul Allaire, Professor of Engineering at UVa and noted expert in magnetic levitation, to design the magnetic bearing system for the EDU. A draft design was submitted to AFS in mid 1995.

Dr. Allaire and his colleagues at the UVa's Department of Mechanical, Aerospace, and Nuclear Engineering were subsequently contracted to design a suitable magnetic bearing test rig. This effort was completed in the fall of 1996.

3.1.2.4 Management and Engineering Support Contractors

From late 1992 to date, AFS has acquired the management and engineering support required to manage EDU development, and provide essential technical expertise. The contracts have ranged from a few days (e.g., short-term requirements for specialized technical skills) to nearly four years for program management support. As discussed elsewhere in this report, the long-term contractors that have supported AFS during EDU development are Dr. Edward Zorzi, Dr. James H. Williams, Dr. Robert B. Bartlett, Mr. John V. Coyner and Mr. Ray F. Griswold. The EDU Program support activities performed by these consultants have included contract negotiation, management and reporting, and technical planning, evaluation and analyses.

3.1.3 Vendor Requirements and Selection Process

The requirements for vendors fall into three categories:

- **Technology Development**. HSSO, ORNL and the University of Virginia were contracted to develop technology suitable for use in the EDU. HSSO, the original lead technology development contractor for the AFS EMFB Program, was principally responsible for the design, procurement, fabrication, and testing of the EDU. ORNL, the original contractor for the fabrication, assembly and initial testing of the EDU flywheel rotor system, has been primarily responsible for completing the EDU design, fabricating the EDU flywheel rotor system assembly, and conducting EDU tests. Dr. Paul Allaire (UVa), the original EDU magnetic bearing design contractor, has been responsible for the magnetic bearing concept analysis and design and the magnetic bearing test rig design.

  These contractors have been primarily responsible for the AFS intellectual property which has emerged from the EDU program to date. Consequently, the selection procedures have been rigorous. In each case, several candidates

were identified having the required technical capabilities, institutional credentials and necessary facilities and resources. For candidate contractors, proposals were requested, reviewed and evaluated. Relevant credentials were verified through site visits and references. For the selected candidates, AFS negotiated the required work scope, costs, schedule, and contract provisions essential to AFS' business position.

- **Suppliers of Equipment and Materials.** Throughout the EDU program acquisition of equipment and materials has been essential to the progress of EDU development. The required materials have included magnetic materials (e.g., rare earth materials), composite materials (e.g., carbon fibers), and structural materials (e.g., marjoram steel). The required equipment included an air turbine for flywheel rotor testing and mandrels for composite windings, among others.

  Unless there is a significant time and cost savings, AFS has typically delegated these responsibilities to its technology development contractors (i.e., HSSO, ORNL, and UVa). In the instances of delegating authority, AFS has invoked its customer privilege of requiring either approval or concurrence with the decisions of the technology development contractor. Most often, AFS relies on the technology development contractor to provide a list of 2-3 supplier candidates and a summary of the experience each has had before AFS will support a procurement decision.

3.1.4 Overview of Honeywell Satellite Systems Operation (HSSO) Program Efforts

The approach for the EDU as detailed in the Honeywell work plan (and as modified by the Honeywell re-plan), included the following major steps:

1. Determine the functional requirements for the EMFB.
2. Develop initial "end-to-end" vehicular / EMFB model to determine component requirements.
3. Establish flow down specifications for all components.
4. Establish the availability of contributory technology resources and preferred subcontractors from technical "due diligence".
5. Provide and evaluate strawman design concepts.
6. Upgrade/refine analytic capability as needed.
7. Complete the EMFB system engineering and design trades to establish system envelope and design parameters.
8. Refine component flow down specifications.
9. Design the instrumented EDU test facility. Fabricate and assemble the EDU test rig.
10. Design and fabricate the system components for EDU integration.
11. Test EDU components as scheduled / available.
12. Evaluate completed EDU.

The initial EMFB EDU approach consisted of parallel component development efforts which would provide verified performance through component tests. These components would then be assembled and integrated into the EDU for further system testing. Design modifications as required based upon EDU test data would then be implemented into a fully functional electric vehicle prototype EMFB. Within five (5) months of the program's commencement Honeywell advised AFS that this undertaking was much more technically challenging than they had originally envisioned. This resulted in a re-evaluation of the development process by the program team and a re-plan by Honeywell of the technical approach. The re-plan was presented to AFS in November of 1993. This re-plan was presented to SMUD in February of 1994 once AFS had reviewed and verified the revised program met the original "spirit and letter" of the EMFB electric vehicle development effort. The re-plan then served as the work plan for the remainder of the Honeywell effort.

3.1.4.1 Introduction to HSSO Efforts

To maximize the technical creativity and streamline the development process, AFS did not constrain HSSO to the EMFB concept per the AFS patent. This permitted HSSO to capitalize on their existing background/technology. It also permitted HSSO complete freedom to fully leverage its creative efforts and develop alternative EMFB configurations. HSSO's technical effort commenced with the EMFB development "Kickoff" meeting in May of 1993 at which time HSSO decided to pursue independent designs for the EMFB with AFS. The schedule and technical approach were as outlined in their detailed work plan, Reference 3-2.

Exemplary leadership communications and technical coordination were absolutely essential for a successful program. To accommodate this need, AFS and Honeywell instituted a weekly teleconference. In addition, HSSO was contractually required to submit monthly progress reports. A number of executive level "Technical Management Reviews (TMR)" and "Program Management Reviews (PMR)" were also conducted, some with Oak Ridge National Laboratory, SMUD and/or ARPA personnel. The more important meetings are summarized in Table 3-2:

Table 3-2 Summary of Program Meetings

Date	Meeting	Comments
Year 1993		
May	"Kickoff" Meeting	Review of work plan
June	Program Management Review #1	Management / technical review
July	Program Management Review #2	Management / technical review
August	Preliminary EDU Specification	Presentation and review of specifications
September	ARPA / SMUD Review	Presentation by AFS of SMUD program
October	Program Management Review #3	Management / technical review
November	Program Management Review #4	Management / technical review
December	EDU Internal Design Review	Full review of initial EDU Design
Year 1994		
January	Program Management Review #5	Management / technical review
February	Technical Management Review	Detailed technical review
May	ARPA / SMUD Meeting in Sacramento	Presentation by AFS of SMUD program
June	Program Management Review #6	Management / technical review
July	ORNL "Kickoff" Meeting at HSSO	ORNL subcontract commencement

3.1.4.2 Technical Challenges / Approach

The components in an EMFB are highly inter-dependent. This is not unusual for most high performance, high speed rotating equipment. The multitude of interrelated technical requirements which challenge the EMFB as a system and the individual components are substantial. Changes, even those that might be considered small or rather unimportant, can cascade into large system alterations. These technology issues and design considerations offer substantial challenges to the engineer and technologist in providing a safe, fully functional and cost effective design. A check list as developed under this effort follows:

1. **Safety:**
   a) Touchdown bearing reliability and life
   b) Safety margins for components and system
   c) Material containment (rotor burst etc.)
   d) Structural interface with electric vehicle (gimballing etc.)
   e) Fault diagnostic strategies / operator interface
   f) Available instrumentation

2. **Flywheel energy storage capacity, specific energy:**
   a) Material selection/properties
   b) Weight / volume constraints
   c) Rotor design / form factor
   d) Rotor spin speed, acceleration / deceleration rates
   e) Composite rim structural integrity / rim - to - rotor hub interfaces
   f) Magnetic / structural materials interface

3. **Long term energy storage:**
   a) Vacuum maintainability
   b) Motor / generator losses and drag
   c) Magnetic bearing losses and drag
   d) Component control system electronics power
   e) System power electronics

4. **Energy conversion efficiency:**
   a) Motor/generator efficiency
   b) Control drive electronics power requirements/electronic efficiency
   c) Charge/discharge rates

5. **Tolerance to mobile base motion disturbance:**
   a) Structural damping and attenuation - isolation sub-systems (gimballing etc.)
   b) Magnetic/back-up bearing performance
   c) Control bandwidth requirements
   d) System dynamics
   e) Sensor selection / performance
   f) Overload protection - touchdown mechanism

6. **Life of System:**
   a) Discharge/recharge cycles
   b) Time at load (stress - rupture)
   c) Imbalance changes over operational life of unit
   d) Reliability of electronics and key components

7. **Cost Factors:**
   a) Design approach
   b) Cost of materials
   c) Manufacture / producibility
   d) Maintenance requirements
   e) Expected life cycle
   f) Salvage value - secondary life

Throughout this program HSSO continued to perform technical "due diligence" efforts to meet contractual requirements in seeking the best technology available from government and commercial resources. HSSO, together with AFS, evaluated numerous organizations for background technology and applicable resources. From that review Honeywell offered "make - or - buy" sub-contract recommendations to AFS. Key technology resources which were included in the evaluation are shown in Table 3-3:

Table 3-3 EMFB Component Suppliers

Company	EMFB Components and / or Technology
• Applied Materials Technologies	composite rotor, housing
• Ashman Consulting Services	motor / generator
• AVCON	magnetic bearings
• Brunswick Defense	composite housing
• Draper Labs	magnetic bearings
• General Corp - Aerojet Sacramento	composite rotor, housing
• Iwon Motor Corp.	motor / generator
• Kingsbury	magnetic bearings
• Lawrence Livermore National Lab	composite rotor
• Magnetic Bearings Inc.	magnetic bearings
• McClellan AFB	composite rotor, housing
• Mechanical Technology Incorporated	magnetic bearings
• Montivideo Technology	motor / generator
• Oak Ridge National Lab	composite rotor, motor / generator
• SatCon	motor / generator, magnetic bearings
• University of Ottawa	composite rotor
• University of Maryland	composite rotor, magnetic bearings

One of the initial requirements identified was to develop the full specification package for the EMFB operating in an electric vehicle. After considerable effort, the more difficult test data for shock and vibration were obtained and subsequently analyzed to provide the vibration/shock spectrum. That data, combined with other information provided by internal and external programmatic resources, resulted in the "technical specification" or program expectations shown in Table 3-4:

Table 3-4 EMFB Program Objectives

Specification	Program Objective
Total Energy per EMFB	3.6 kWhr
Peak/Continuous Power per EMFB	16 kW / 8 kW
System Weight	40 - 50 kg (est .)
System Volume	56L (est.)
Specific Energy	65 - 80 Whr/kg
Specific Power	320 - 480 W/kg
Energy Density	64 Wh / L
Power Density	286 W/L
Active Cooling	None required
Efficiency	>92% as a battery (round trip)
Recharge Time (50% / 100% recharge)	< 14 min. / < 24 min.
Energy Storage Time	> 10 days
Years of Operation in Electric Vehicle	12 yrs.
Acoustic Signature	< 50 dB at 1 meter
Electrical Bus	220 - 440 vdc
Vibration - Normal	± 1 g over full frequency range
Vibration - Severe	± 2 g over full frequency range
Shock - Normal	± 0.5g
Shock - Transient	± 2g
Ambient Temperature Range of Operation	- 30^0 C to + 70^0 C
Operational Humidity Range of EMFB	0% to 100% RH

3.1.4.2.1 EMFB System Considerations

Consideration (1) - System Configuration and Interdependence of Components

To deal with the interdependence of design and operational issues a full optimization analytic software package was implemented by HSSO. The software predicted general design, weight and envelope requirements for a given energy/power requirement. The optimization offered preliminary performance assessment of the EMFB including component design parameters for the; composite rotor, motor/generator, magnetic suspension, touchdown mechanism, housing, and later in the program some provisions for electronics. This capability proved extremely effective as AFS and HSSO reviewed potential designs for electric vehicle application ranging from 2 kWhr to 14 kWhr in size. The simulation model was continuously updated as information and component refinements became available. The 3.6 kWhr design was selected by AFS based upon data supplied by HSSO from that analytic modeling.

Consideration (2) - Determination of Vehicular Requirements

Although the automotive gimballing design/development was not an explicit part of the American Flywheel System contract with Honeywell, it is a key issue for suspension isolation and dynamics. Interaction between the EMFB suspension system (magnetic and /or touchdown) is a system consideration not overlooked in this effort. Trades between gimballing requirements and system performance were incorporated into dynamic simulations for the electric vehicle. This information was integrated into HSSO EMFB design considerations and system evaluations. Various scenarios were evaluated (e.g. figure 3-2) using a special dynamics simulation of the electric vehicle undergoing shock and vibration disturbances. The shock / vibration criterion were obtained from public domain sources, the best available to HSSO at that time. The vehicular loads as determined for the analysis were used as a basis to evaluate the rotor-bearing system loading and dynamics of the system.

Consideration (3) - Thermal Management

Thermal issues are important factors when considering thermal management for components encapsulated in a vacuum chamber. Convection, as generally depended upon in many rotating devices will not offer the remedies typically available, as the vacuum enclosure prohibited that luxury. Special conduction paths were established where required to deal with thermal stability and control. Of particular relevance was the motor/generator. Here special design/assembly techniques were developed to minimize the system losses as well as extract heat. These are the subject of active AFS patent disclosures.

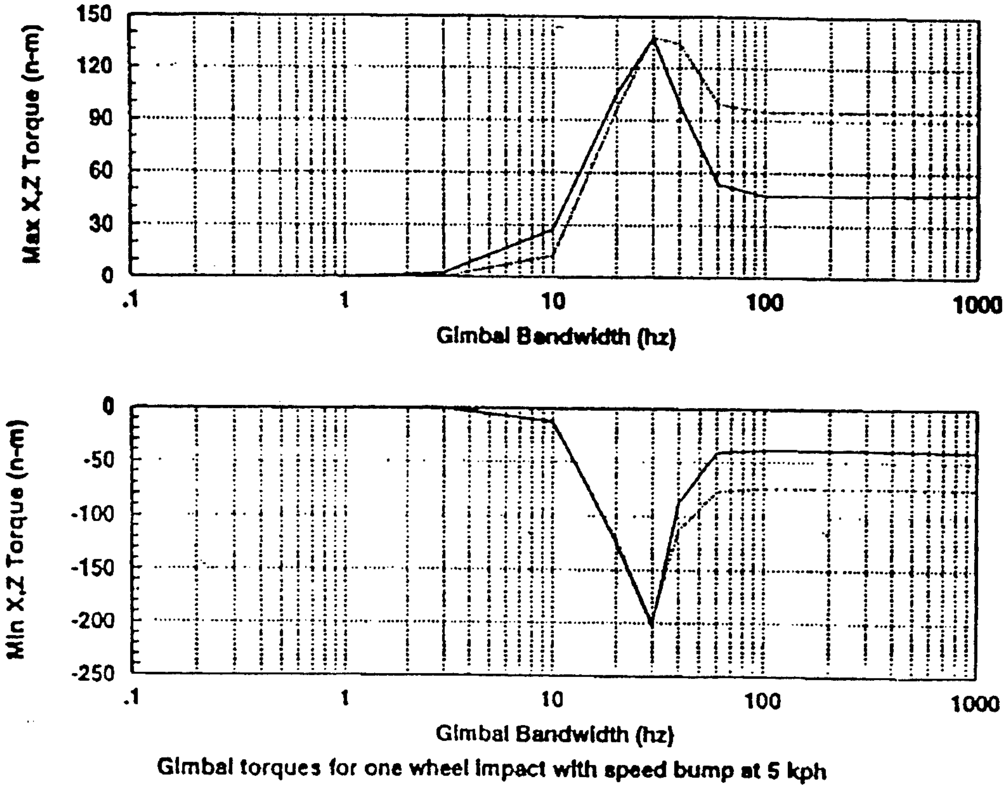


Figure 3-2 Simulation of Vehicular Loads for Speed Bump

Consideration (4) - High Speed Rotor Dynamics

Rotor dynamics for the entire EMFB EDU system is a key design factor for developing safe and reliable high speed rotating machinery. Issues such as critical speeds, stability and imbalance response are pivotal to mature design approaches. A special finite element based rotor dynamics resource was implemented throughout the program by HSSO (and later by ORNL) for the EDU development. Changes and alterations were incorporated into this simulation which included provisions for the composite rotor, suspension system, motor/generator, housing and accompanied structure. The evaluation of the rotor's ability to dissipate unwanted perturbations arising from road conditions or vehicular motions was incorporated. Critical speeds and imbalance sensitivity were factors of importance included in these evaluations. Special attention was given to dampening structural modes in the operating range. The rotor was designed to eliminate any modes which couldn't be controlled through bearing damping.

Consideration (5) - Limitation of Available Information on Containment and/or Burst Testing of Composite Rotors

Although the development of composite rotors for flywheel applications had been unsuccessfully attempted in the 1970's resulting from the "gas embargo", few publications documenting burst tests or containment evaluations are available. One of the few exceptions is the testing performed at ORNL in the late 1980's. This intentional burst test, which set the world's record for specific energy of the rotor at 117 Whr/lb, offered a technical foundation for further evaluation. Photographs and recollection of key individuals offered HSSO and AFS the opportunity to reverse engineer a basis for insight, (although limited), into the forces associated with a burst composite rotor. (Note: ARPA has since lead further efforts to establish safe containment and rotor design criterion. ORNL is developing their data in a more rigorous manner for the purposes of future dissemination).

Consideration (6) - Material Cost

High strength composite materials as envisioned for application in the composite rotor are not inexpensive. This is also true for some of the rare earth magnetic materials for the magnetic suspension and motor/generator. Cost factors will obviously improve as production quantities increase, but careful attention to the deployment and introduction of the higher performance/higher cost materials is a challenging assignment for the development of the EDU and EMFB. Cost issues, including manufacturing processes must be kept in perspective as this program proceeds.

Consideration (7) - System Efficiency and Losses

Of notable importance to system efficiency and thermal stability is system losses. The EMFB was designed to maximize the energy storage time. This means that losses from all components are held to an absolute minimum. Electro-magnetic eddy current, material and losses from stray magnetic flux are of significant concern. The design approach for the magnetic bearings and motor/generator are particularly sensitive to this issue as discussed later herein. Windage losses from outgassing of components or vacuum leaks are another source of undesired energy drain. Although the vacuum integrity of the housing was to be addressed in detail in the prototype (post EDU) stage of the program, it is also addressed from the prospective of providing an adequate EDU test facility.

Consideration (8) - Extended Vacuum Operation of Composite Materials

As a final system consideration, material outgassing characteristics for the composite materials to be utilized on the numerous EDU components, is of significance. Accordingly, tests were performed at Lockheed Missiles and Space Company Inc. in Sunnyvale California for HSSO to determine the suitability of these composite fiber/matrix combinations for extended vacuum operation. Samples were provided which consisted of three fiber/resin combinations. Quantitative characterization of the outgassing rates of these materials was determined per ASTM E-1559 "Standard Test Method for Contamination Outgassing Characteristics of Spacecraft Materials". The outgassing flux from the sample leaves the effusion cell and is monitored by quartz crystal microbalances held at differing temperatures. The test chamber as utilized per figure 3-3 below

maintained a pressure of 10^{-9} Torr for the test sequence. Figure 3-4 illustrates typical data as obtained from these outgassing tests and evaluations by Lockheed.

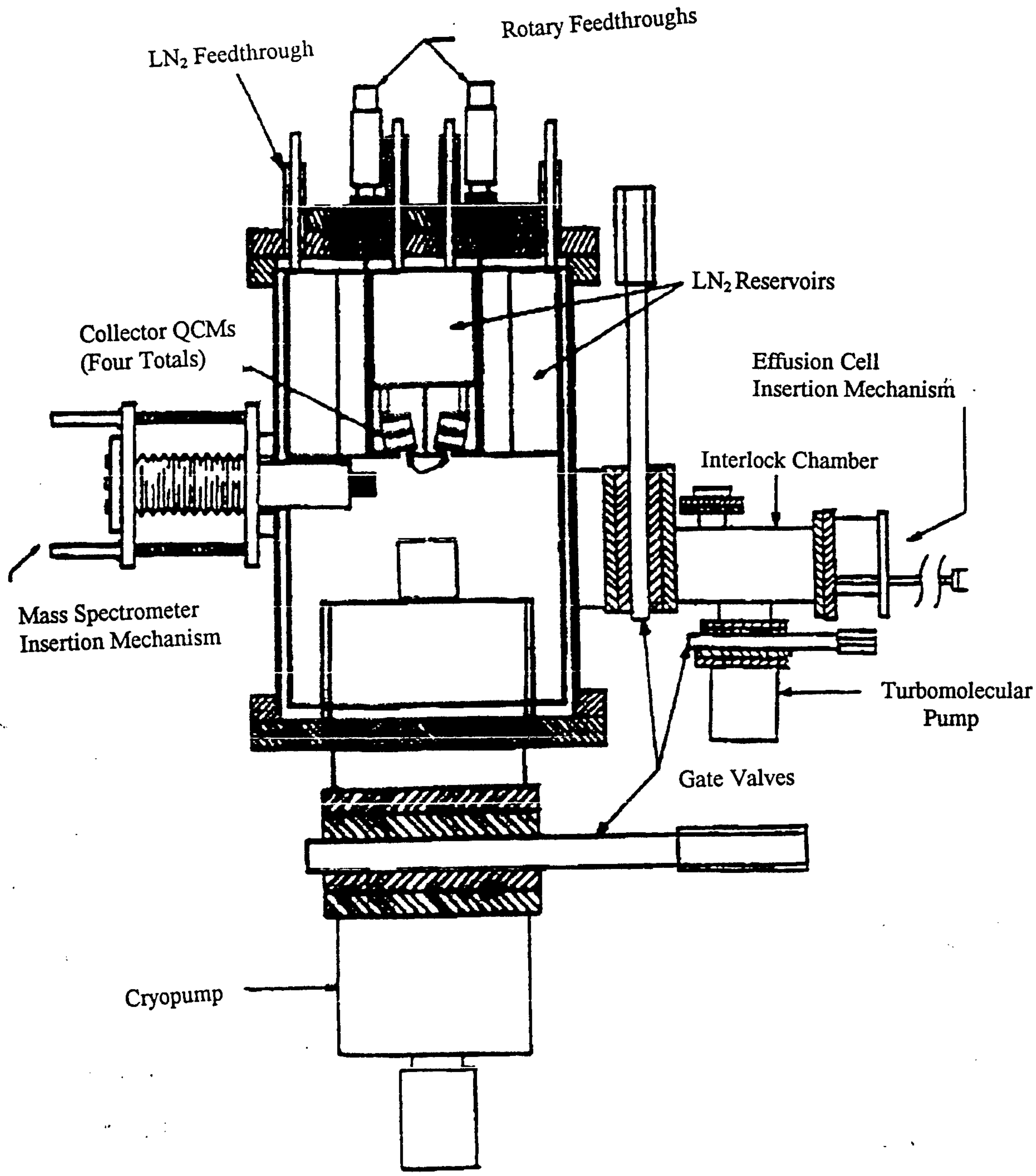


Figure 3-3 Material Outgassing Testing Apparatus

3.1.4.2.2 EMFB Rotor Suspension and Bearings

3.1.4.2.2.1 General Considerations

The first generation EMFB EDU suspension components as developed by HSSO for AFS are unique in design and are expected to perform exceptional when they undergo testing.

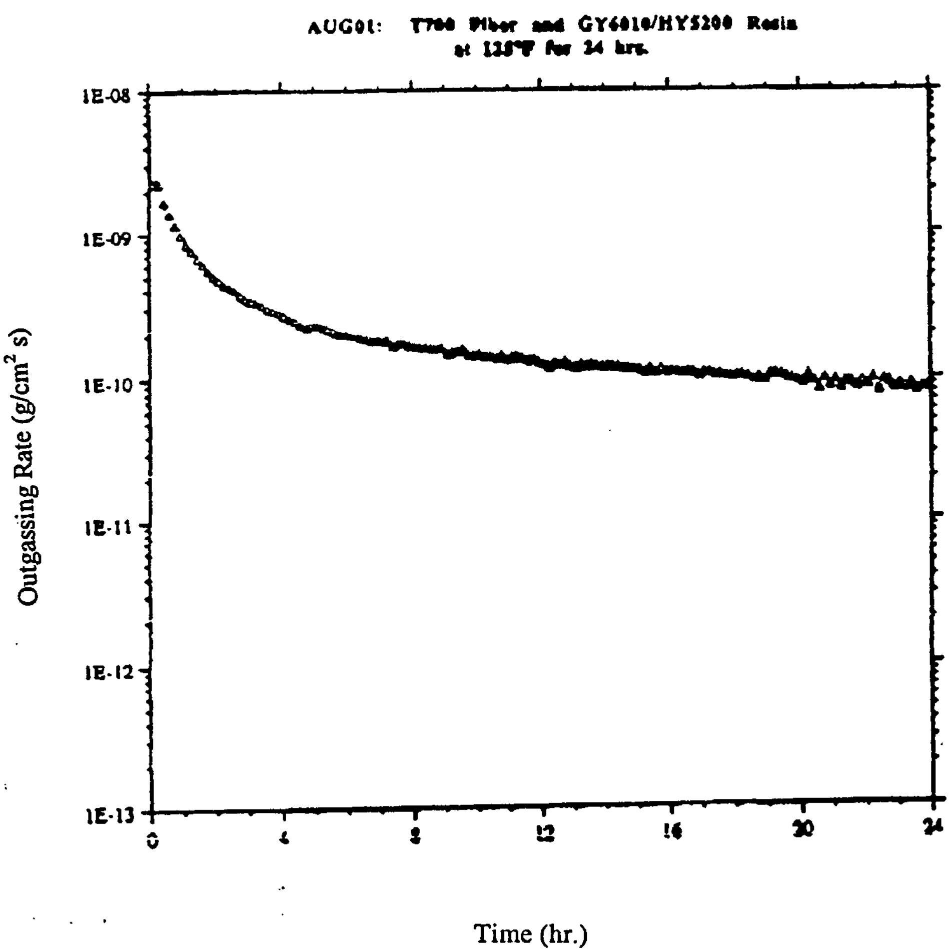


Figure 3-4 Typical Material Outgassing Test Results

The EMFB is installed in a vertical orientation offering limited gravity loading on the radial magnetic bearings. However, off vertical operation of the EMFB, even in a quiescent (stand-by) state, offers minimal loading conditions. The radial magnetic bearing must support the rotor under the conditions of: (a) quiescent operation, and (b) normal road/shock vibration for the electric vehicle. Under severe shock or crash conditions a set of touchdown (back-up) bearings assist the magnetic suspension in support of the rotor. All efforts to minimize weight, power losses, and geometric envelope yet provide adequate stiffness and damping for acceptable rotor stability was deemed essential. The activities for the EDU suspension development to be performed by HSSO included:

1. Determine the functional requirements of the complete EMFB suspension system in an automotive environment
2. Complete the EMFB system engineering and bearing design trades to establish system envelope and design parameters
3. Design the EDU suspension's mechanical and / or electronic parts, and
4. Fabricate the EDU's suspension for the integration into the EDU.

All but the integration of the hardware into the EDU were completed by Honeywell prior to cessation of their efforts under this program. The alternate design magnetic bearing activities continued under AFS contract to Dr. Paul Allaire (UVa) as described in Section 3.1.6 below.

3.1.4.2.2.2 Touchdown Bearings / Mechanism (Back-up)

The touchdown bearing must endure a aggressive and hostile environment due to its primary role of providing support to the rotor system under high loss/shock or adverse conditions. Vehicle motions can be severe at the lower frequencies which further challenge the integrity of the backup bearings. The question of which, or what type of bearing would be adequate, was initially assessed by HSSO. Key considerations in the design and development of the touchdown bearings and mechanism were:

1. Cost and weight of a more actively utilized touchdown system vs. the additional cost, complexity and weight and the magnetic suspension.
2. Cost and durability of journal bearings, non- lubricated designs, low cost replaceable designs as compared to rolling element bearings.
3. The impact on overall EMFB system efficiency, specific energy etc., and customer acceptance of the selected design.

From the AFS/HSSO evaluation it was determined that the best approach would be to incorporate a mechanical rolling-element bearing, or more specific, an angular contact ball bearing. This assessment ruled out other types of bearings for solid technical reasons but left the key question of materials (balls, races, separator, lubricant etc.) to be empirically determined. It was felt that this scenario would be the best approach for the EMFB touchdown bearing development.

A tribological material screening process under simulated simultaneous combinations of pressure and velocity required a special test rig/fixture, Figure 3-5. The

fixture tests pin - on - disk wear profiles to 15 million psi-fpm. Twenty eight (28) test combinations of pin material, disk material, sliding velocity and lubricant were evaluated.

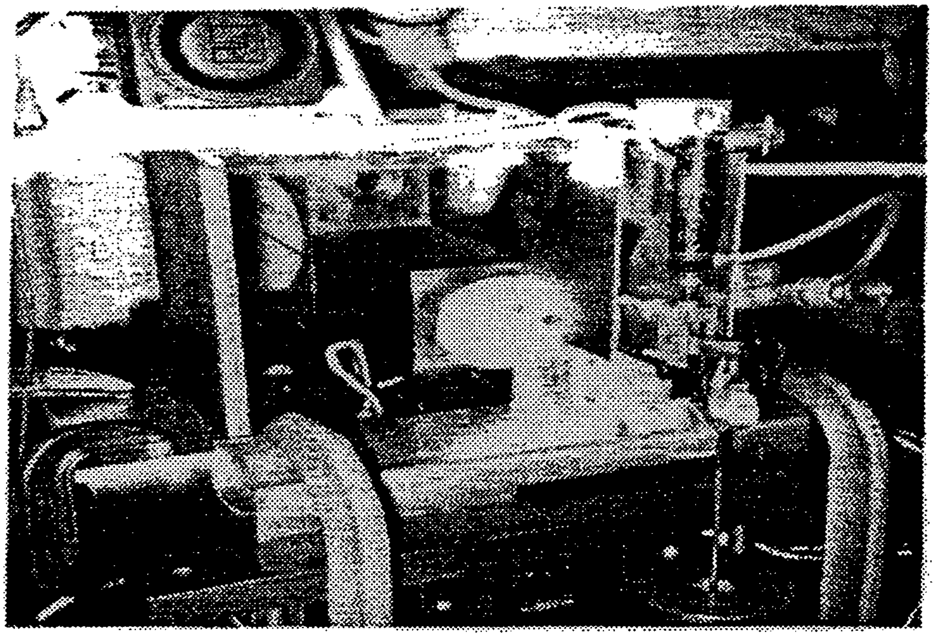


Figure 3-5 Pin-on Disk Test Apparatus

Test materials (pin/disk combinations) as selected for evaluation included per Table 3-5:

Table 3-5 Pin-on Disk Material

Pin / Disk Material
• Stellite 6B / Stellite 6B
• Stellite 6B / 440C
• Alumina / Zirconia
• Zirconia / Alumina
• M50 / Stellite 6B
• 440C / Stellite 6B
• 440C / Alumina
• M50 / 440C
• 440C / 440C
• Nitronic 60 / 440C
• Stellite 6B / Nitronic 60

Results clearly indicated superior combinations (per Figure 3-6) were available to AFS for possible implementation into the touchdown bearing and mechanism. A configuration using reasonable cost, least wear material and lubricant combination was selected to be implemented in the containment vacuum environment for an electric vehicle.

3.1.4.2.2.3 Magnetic Suspension System

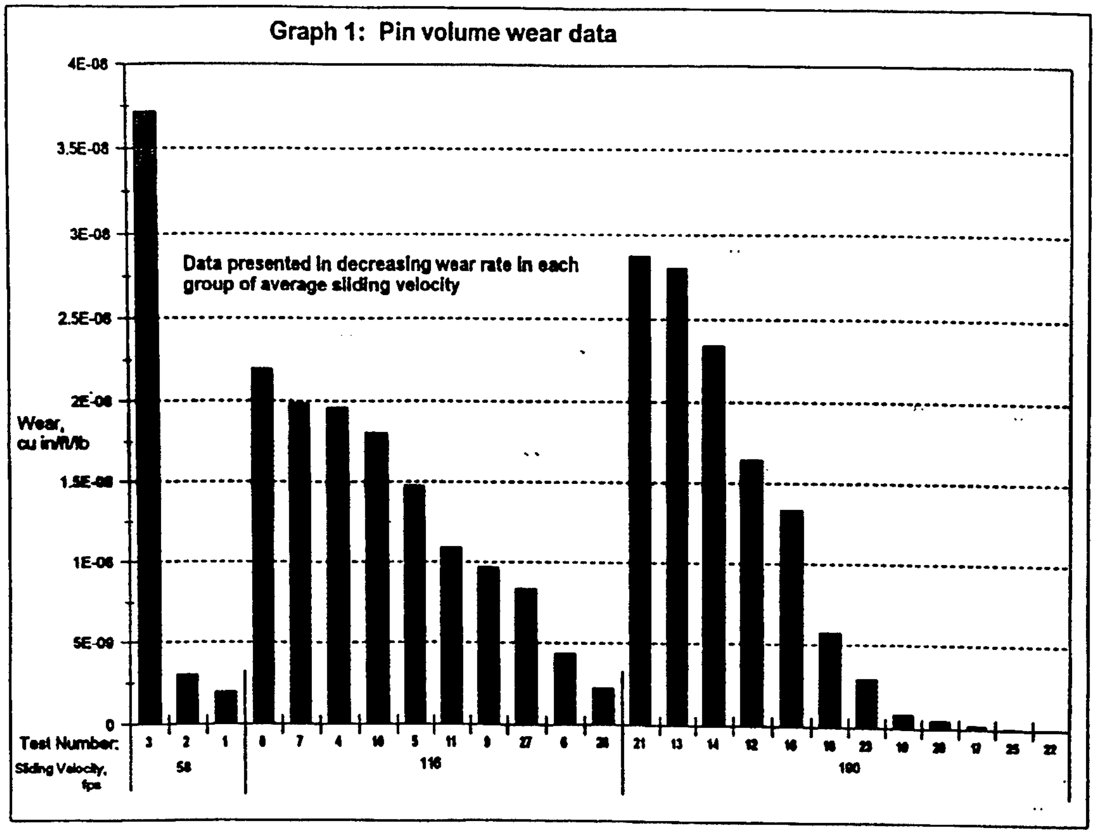


Figure 3-6 Pin-On-Disk Wear Data

The automotive application requires that the mechanical battery operate on all common road surfaces. Consequently, possession of such data as to specific road conditions is important to determining the design conditions and expected attributes of the rotor suspension. HSSO and AFS found that the Eaton corporation had performed testing for the Department of Energy (DOE) to determine battery life under vibration. They were very cooperative in supplying the DOE report and supplemental verbal data to HSSO and AFS. This data combined with Society of Automotive Engineers (SAE) data, Reference 3-5, and per Figure 3-7, provided needed and necessary data to determine realistic shock and vibration characteristics for the AFS program. This data was also used to simulate electric vehicle "disturbances" for simulation and analyses of the EMFB under specific road conditions as exemplified per Figure 3-8.

The magnetic bearings must levitate the EMFB flywheel rotor throughout the normal operating range of the vehicle under all "normally encountered" road conditions.

Due to the high rotational speeds of operation, magnetic radial and axial bearings were the desired approach. As the EMFB rotor must rotate at high speeds, other bearings types, if implemented, could have caused excessive drag (friction), thermal problems, or have been life limiting to the EMFB's operation. Magnetic bearings can be passive, relying simply on magnetic fields generated by magnetic materials, or active, where magnetic fields are generated by an electric current. The bearings must support the rotating components both axially, or along the spin axis, and radial from the spin axis. Selection and trade between passive and active radial and axial magnetic bearings is dependent on the operational parameters being optimized for a given application. Important considerations for the selection of the bearing architecture include; bearing stiffness requirements, drag torque, quiescent operating power, weight, magnetic gap requirements for base motion and runouts, magnitude of automotive motion disturbances and degree of design flexibility. The fundamental activities for the magnetic suspension to be performed by HSSO included:

1. Determination of the functional requirements of the EMFB magnetic suspension for an automotive application.
2. Completion of the EMFB system engineering and magnetic suspension design trades to establish the system envelope and key parameters.
3. Designing the EDU magnetic suspension mechanical and electronic components, and
4. Fabrication of the EDU magnetic suspension for testing.

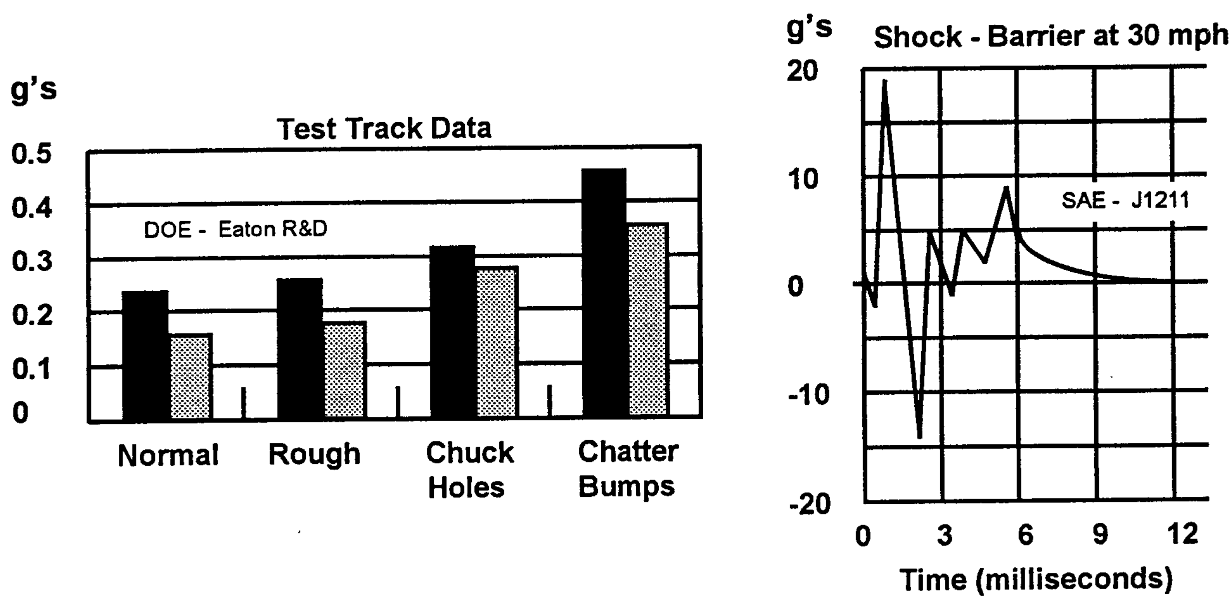


Figure 3-7 Vehicular Shock and Impact Data

Initially calculations were performed to determine the basic requirements for the magnetic bearings. In the fall of 1993 HSSO performed a material survey to determine the structural, magnetic and related material properties which could be made available for this effort. Specific attention was focused upon the structural capabilities of rare earth

magnetic materials. Additional due diligence of possible vendor and suppliers convinced HSSO that the correct approach to this development effort was to perform this effort in house. A component specification was then prepared consistent with the EDU design and as provided earlier per Section 3.1.4.1. of this report. Sizing and preliminary analyses were then applied to the vehicular dynamic simulations to determine interactions of major components (e.g. rotor, motor/generator, magnetic suspension, touchdown mechanism etc.). These were incorporated into the system dynamic analytic models as well as into the EMFB optimization software. As refinement of the design became more challenging the use of 3-Dimensional Boundary Element Models (3-D BEM) was required. The 3-D BEM provided detailed flux path assessment offering the possibility of minimizing stray flux and thereby minimizing unneeded losses and thermal problems within the bearings, Figure 3-9.

Of particular importance in the design trades for the magnetic suspension is the issue of suspension dynamics for safe and reliable operation of the EMFB. These factors must be incorporated into the EDU's bearing configuration to permit early assessment of bearing operation. Critical speeds, stability and imbalance sensitivity were assessed incorporating specialized finite element resource developed by the Electric Power Research Institute (EPRI) called FEATURE. It was made available for this effort by license through Mechanical Technology Incorporated and was applied throughout the suspension development cycle. In addition, structural dynamics resources were also supplied to evaluate the dynamics (natural frequencies, mode shapes) of key components.

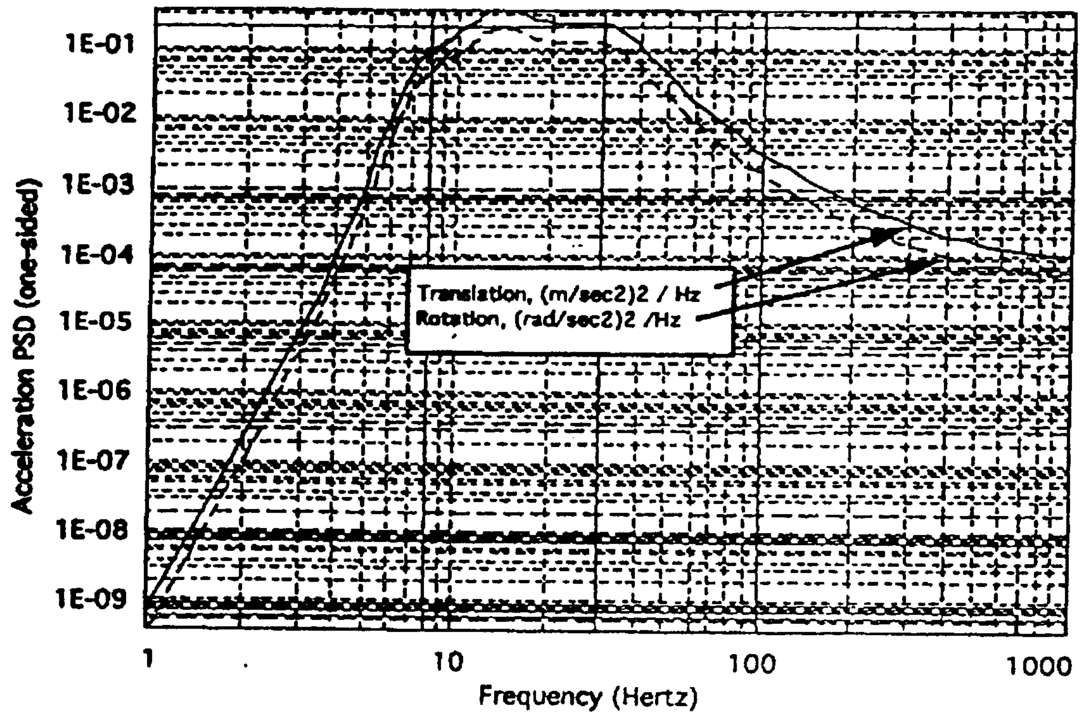


Figure 3-8 Vehicular Base Motion - Frequency Spectrum

The HSSO passive radial design configuration incorporates several advantages. It provides radial stiffness without consuming any electrical power. The flux path is constant and uniform while rotating normally, so it should not produce any eddy current losses. the only source of changing flux density would be from the mechanical imperfections in the manufacture of parts. The radial stiffness per pound is excellent, providing a minimum weight design. The only disadvantage is that it offers increased loading in the axial direction, but this has some redeeming facets. It provides the opportunity to operate the rotating assemble slightly above center which will bias in opposition to gravity. This will provide a passive one "G" bias to suspend the rotating parts without the use of constant electrical power to provide levitation. The axial loads are compensated by the thrust bearing.

The HSSO active radial bearing is a unique design that uses both permanent magnets and electro-magnetic fields. The permanent magnets produce a constant and uniform flux in the air gaps. Coils located in opposing pairs are stationary. When current flows through the coils they are either drawn toward the center of the rotor or are repelled from it depending upon the direction of current flow. The coils are operated in pairs so that their forces are additive in the radial direction. The only flux variation in the pole material is from the change in the air gap reluctance caused by energizing the coils. This is only present when the coils are being activated and is minimized by the low ratio of ac flux to dc flux. This high response is required as it is the primary function of these actuators to

dampen the lateral displacement of the rotor. The high bandwidth is advantageous to maintain rotor stability and frequency control of lateral shaft dynamics.

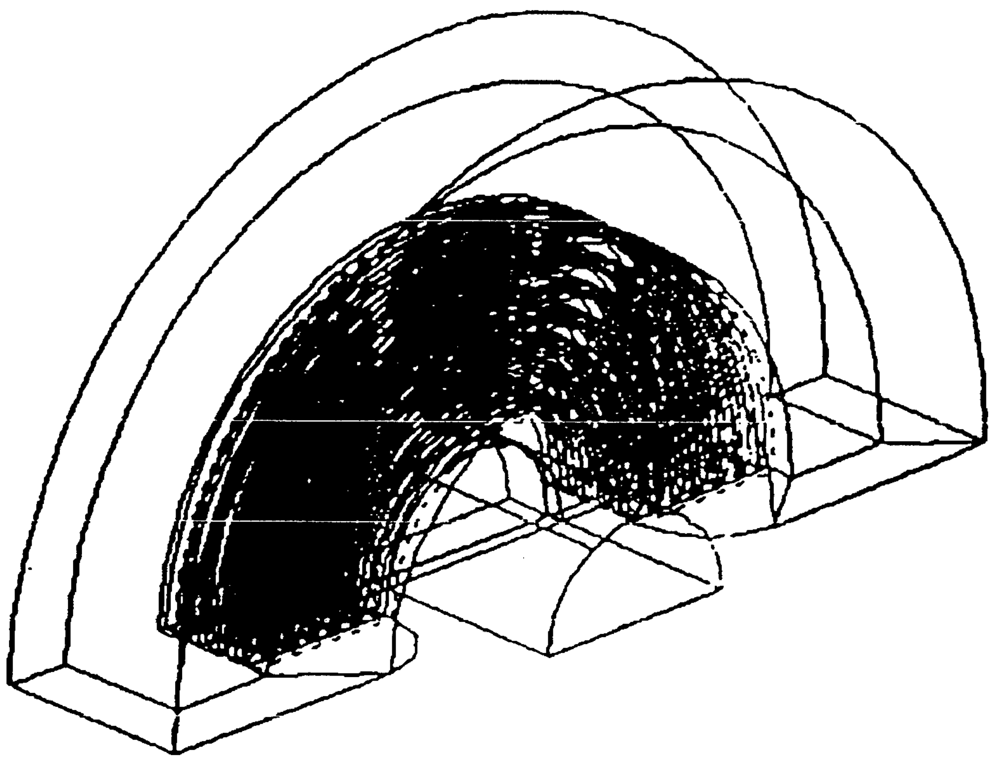


Figure 3- 9 Magnetic Bearing Flux Map (3-D BEM)

The first generation magnetic bearing design was developed at HSSO for the radial and thrust bearings. This effort was continued to in alternative design per Section 3.1.6.

3.1.4.2.3 Vacuum / Safety Containment for the EDU

The containment/vacuum housing design and development task was limited under the EDU effort to that which was required to satisfy the EDU test facility requirements and offer insight into the prototype design. The EDU housing was not viewed as identical to the final prototype design but was designed for extensive instrumented testing of the components, sub-assemblies and system. However, the EDU containment vessel is a safety enclosure which must be capable of restraining particulate and solid debris resulting from a flywheel or component failure. The test cell would be expected to further assist the containment housing by providing a supplementary safety system for the EDU rig. The specific HSSO EDU containment design is as illustrated per figure 3-10 and consists of the outer cylindrical portions and end caps. The inner most metallic liner is primarily a vacuum enclosure, but one which is suitable for assisting thermal transfer from debris impact. The middle drum assembly is a composite, Kevlar, which is capable of providing structural integrity to the EDU during a failure. Kevlar became the material of choice due

to its excellent reputation in bullet proof shields used by law enforcement agencies, its general availability, and its reasonable cost. As shown, the spin chamber also includes a heavy steel outer wrap for added safety.

A sustained vacuum level of 10^{-4} to 10^{-5} Torr. requires special considerations for sealing. Typically a vacuum housing for this type of application consists of two or more housing sections joined together and incorporating O-ring seals at the joints to prevent leakage of gases into the housing. Typical sealing designs would permit substantial leakage and the vacuum integrity would be violated with a few days, Figure 3-11. Permanent closures such as by welding were discarded as impractical. Getter, a substance placed into a vacuum to remove traces of a free gas, was also considered by HSSO for removal of unwanted gaseous elements. Additional vacuum pumps as required were viewed as acceptable for the EDU rig testing, but would certainly offer an unacceptable weight penalty for the Prototype EMFB. To gather needed data on the vacuum integrity it was decided to use similar menthols for sealing the EDU and EMFB prototype. To minimize windage losses due to leakage, multiple O-ring configurations were designed and evaluated.

From a conservative design point of view, and to insure that safety is not compromised, the design of the prototype containment vessel was required by AFS to withstand a full speed rotor burst or failure. It was recognized that this approach may not be compatible with that taken by other developers within the field. However, technical trade space as to probability of failure vs. penalty of failure was not available for the new materials being considered by AFS for the rotor and components. To provided a technically sound basis for the design, the acquisition of empirical data (i.e. intentional burst and failure testing) for the specific design in question is essential. As the AFS composite rotor at ORNL was not scheduled for burst testing in time to support the EDU design effort, the limited data available from public sources and consultants was even more important to this program. Upon request from AFS, HSSO began to develop scenarios for determining loads to be experienced from a catastrophic structural failure. With the full cooperation for the Oak Ridge personnel and leadership, available data offered some insight into the magnitude of loading which might be expected from an intentional full speed burst of the AFS composite rotor. By reverse engineering the data available from the ORNL rotor burst testing of the late 1980's, order of magnitude estimates were developed. Preliminary indications were that the EDU containment together with additional safety measures would be adequate for EDU testing.

maintained a pressure of 10^{-9} Torr for the test sequence. Figure 3-4 illustrates typical data as obtained from these outgassing tests and evaluations by Lockheed.

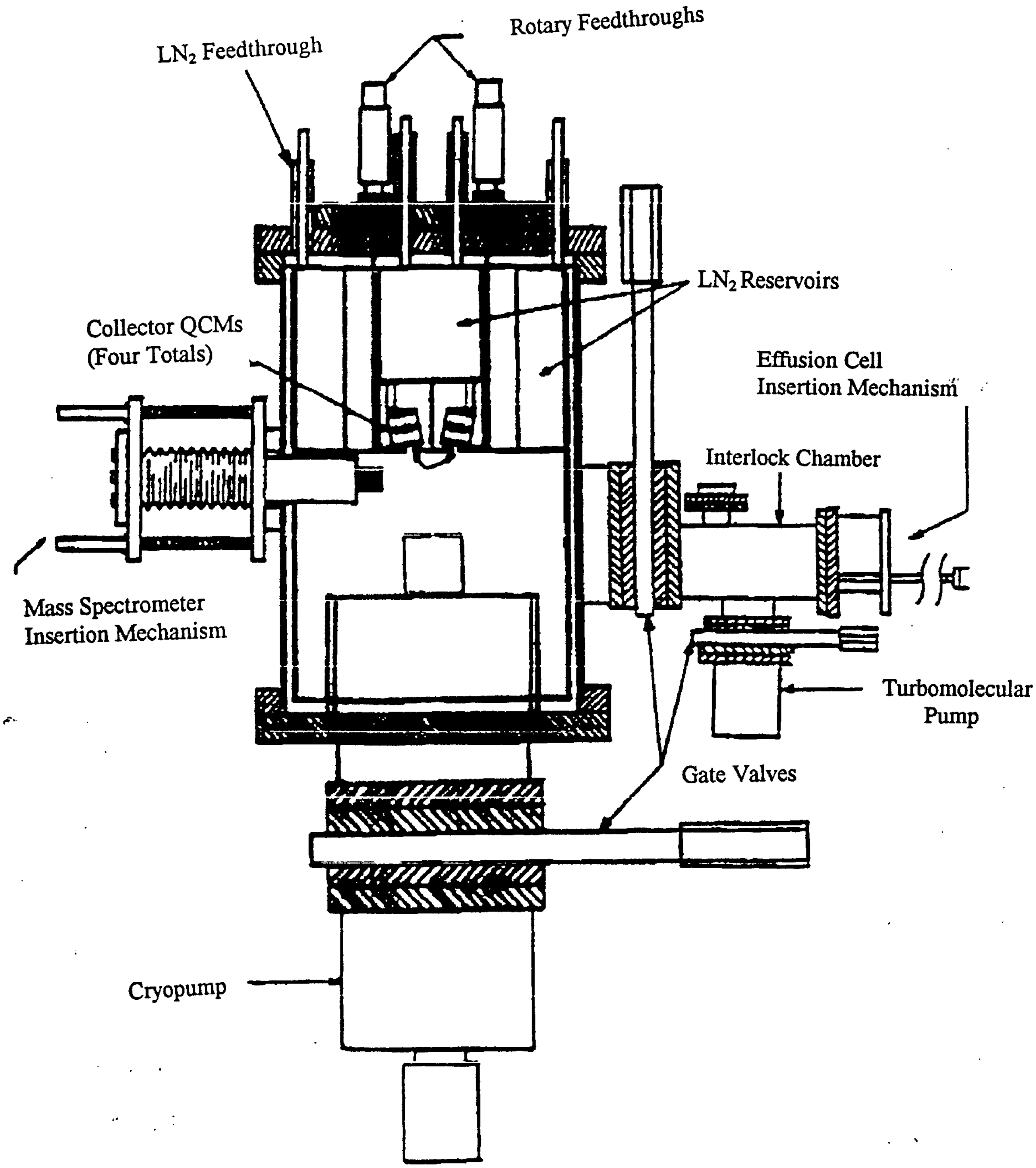


Figure 3-3 Material Outgassing Testing Apparatus

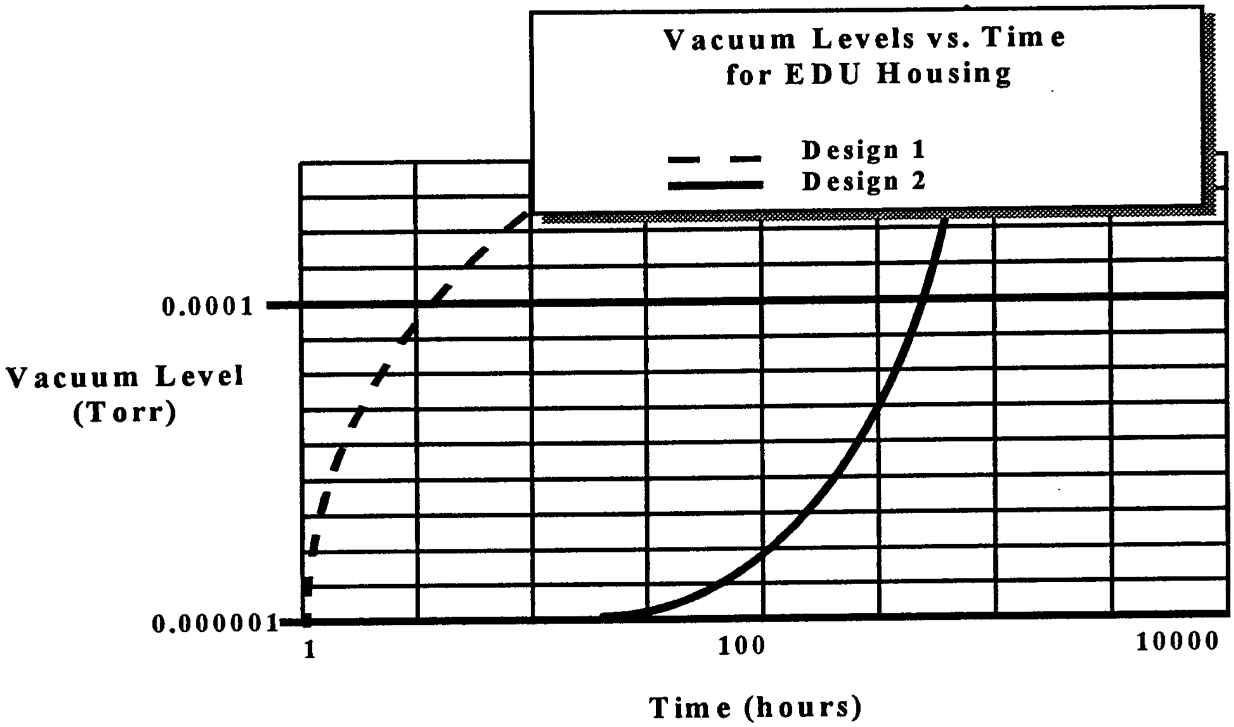


Figure 3-11 EDU Vacuum Levels vs. Time

3.1.4.2.4 Power and Control Electronics

The power and control electronics requirements were established from the automobile; (a) when driving, (b) under acceleration, (c) for regenerative braking, and (d) when parking or at rest. In addition, the requirements were derived from the master dynamic simulation which includes all of the control functions for the power system and the electro-magnetic bearings. The disturbance functions in the simulation are derived from the road test data obtained from the SAE and Eaton Corp. Since the EMFB's will be used in pairs to cancel gyroscopic effects, (cancellation of gyroscopic loads by counter-rotating pairs is discussed in the AFS patent) the power/speed controls must be accurately paralleled, by cross linking the controls of two matching units.

The power electronics are designed so that the EMFB will appear as a normal chemical battery to external interfaces. The electronics will automatically correct for the generator voltage decrease due to decreasing wheel speed to maintain a constant output voltage to the load. It will also take a constant charging supply voltage and drive the spin motor with a constant current for optimum spin up. The baseline design will support the standard vehicle bus voltages of 96 vdc to 240 vdc per Figure 3-12.

The magnetic bearings consist of one axial and two sets of radial bearings. The EMFB is normally operated with the spin axis vertical. The axial bearing supports all changes in "g" loads in this axis. A permanent magnet supplies the nominal one "g" bias. The

controls put the axial bearing into an alert stand-by mode when the vehicle is parked, thereby saving power. Most of the disturbances from road bumps result in vertical accelerations which are sensed by accelerometers and position transducers, and controlled by changing the current in the axial bearing coils. This assures that the rotating equipment is always centered vertically in the assembly. Several advanced control techniques are used in this system to optimize control action while minimizing the power consumption.

The radial magnetic bearings are located near each end of the spinning assembly. They provide stabilization in the radial directions, and also stabilize angular disturbances about any horizontal axis. To accomplish this the bearings are made into segments which are individually controlled. The upper and lower segments on the same side of the spin axis work together to oppose lateral motions of the rotating assembly. Angular disturbances are controlled by activating segments on opposite sides to produce a couple which opposes the disturbance. As in the axial control, both position and angular rate sensors are required for inputs to this control. Several sensors are used which are evenly spaced around the rotating assembly in a plane perpendicular to the spin axis and at each end of the rotating assembly. The controller accepts the signals from the sensors and generates the proper control currents for the appropriate magnet segments to maintain the spinning assembly centered in the non-spinning assembly. Again, advanced control techniques are employed to optimize the control accuracy and minimize the power consumption.

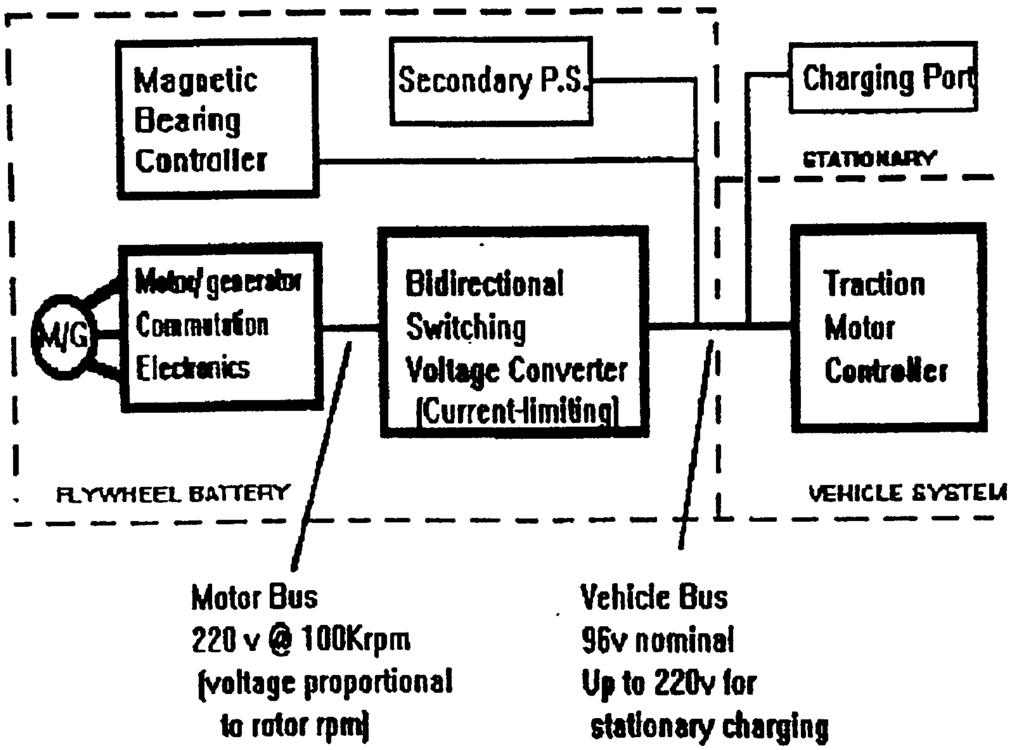


Figure 3-12 Power Electronics System Configuration

3.1.4.2.5 Composite Rotor

A critical component to the EDU is the composite rotor which stores the majority of the kinetic energy. During the energy crisis of the mid 1970s, the predecessor to the US Department of Energy, the US Energy and Research Development Agency funded many prestigious corporations to develop flywheel technology. Many of those involved fell far short of desired performance and specific energy capabilities. To establish the technology capability and resources available to us in the 90s, Honeywell evaluated numerous possible suppliers of the design and manufacture of composite rotors. The specific companies HSSO and AFS assessed are listed in Table 3-3. After reviewing specific proposals from selected suppliers, Honeywell recommended to AFS that Oak Ridge National Laboratory design, develop, fabricate and test the composite rotor. It was argued that; (a) in the mid 1980s Oak Ridge National Laboratory set a world's record for the performance of a composite flywheel rotor at 117 Whr/lb for the flywheel rim which was still valid, (b) ORNL had extensive test facilities which could be dedicated to this effort, and (c) experienced and qualified personnel were available. AFS concurred with the HSSO recommendation and a contract with ORNL Composite Manufacturing Center resulted.

Detailed specifications for the fiber composite matrix for the rotor resulted from the HSSO system evaluations and preliminary NASTRAN structural finite element modeling. From that effort, and consistent with the system specifications as set forth for the program, Table 3-4, ORNL was provided guidelines for the development of the rotor. System integration of the rotor remained HSSO responsibility. ORNL contract efforts are as documented per Section 3.1.5 below.

3.1.4.2.6 Motor / Generator

Based upon HSSO's initial technical evaluation, it became apparent that the most critical issues with respect to the motor generator are the; (1) weight to power ratio requirements, (2) motor and generator efficiency, (3) high rotational speeds, and (4) heat removal when operating in a vacuum. All of these issues needed to be in proper balance for an effective motor/generator design. Unfortunately this was not so straightforward or simple a task. The weight to power ratio dictated a very compact design which unfortunately exacerbated the heating issue, and heating requirements must be kept to a minimum. With the limitation of available light weight materials, high flux density was required wherever possible in the design. This compromised the issue of minimizing stray flux or local eddy current generators, with the resulting negative impact on heat and inefficiency. The high rotational speeds needed for energy storage also produce exceptionally high centrifugal stresses on the materials. Magnetic materials are not the most competent structural members and may require special support or assistance thereby limiting design options. In fact AFS determined that published data for the structural limitations of certain rare earth magnets was in error. Strength testing by HSSO confirmed AFS's insight. Typically, motors of this type generally produce linear torque with input electrical current and independent of rotational speed. The coil current is a major source of heat in the motor and the heat rate is proportional to the square of the current multiplied by the coil resistance. Constant torque must supply the minimal power even at minimum speed. Energy losses must be minimum and the efficiency as a motor and as a generator (charging and discharging power) must be in balance. The challenges were formidable.

Consistent with the approach for other components, the due diligence evaluation commenced once the basic features and requirements of the motor/generator had been determined. That AFS/HSSO assessment was focused upon determining if there were any motor/generators available or any laboratories available in the industry capable of designing and building motor/generators with performance approaching AFS specifications. Clearly AFS did not want to reinvent what had already been accomplished at another available source. The specific suppliers contacted are per Table 3-3. In addition, the resources of Honeywell's Electro-Components in Durham NC were assessed for compliance with the program needs. From its review and initial technical assessment, Honeywell determined that the Durham operation had the technology, capability and experience in the size class which AFS required. In addition, Honeywell Space activities in Arizona had been performing some relevant magnetic research and could offer talented

staff and leadership to the motor / generator development activity. Accordingly Honeywell decided to develop the motor generator using Honeywell's internal resources.

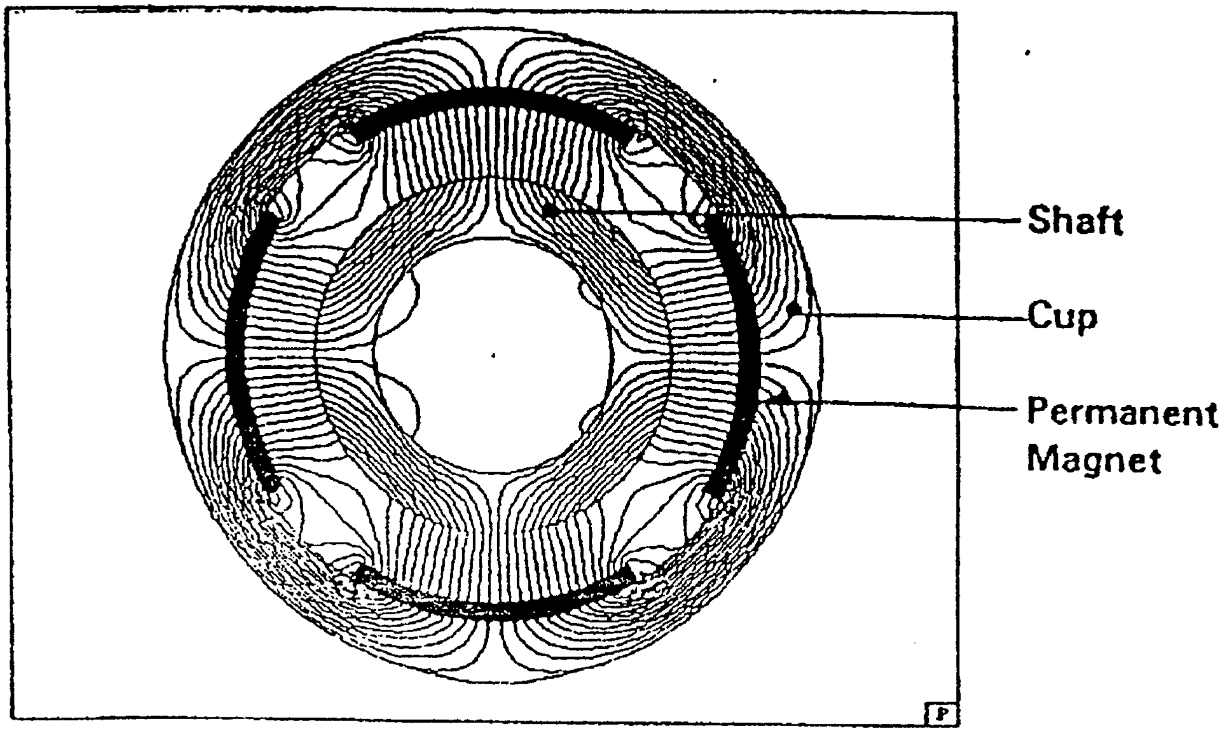


Figure 3-13 Motor / Generator 3-D Flux Field

The preferred configuration was determined and material trade studies concluded. In depth structural evaluation, employing both two and three dimensional analytic methods were utilized. The three dimensional magnetic models were used to configure the final design, Figure 3-13. This capability was complemented by a motor / generator component specific parameter evaluation. The lumped parameter analysis included inputs for speed, power, efficiency, bus voltage, diameters, mechanical clearances, number of poles, motor phase, various magnetic circuit parameters, and material properties including; mechanical properties, factors of safety, magnetic properties, electrical properties, and thermal properties. With the appropriate tools and resources in place the following areas were evaluated; performance parameters, thermal parameters, and normalized drag trends. Naturally many design iterations were needed to meet all of the specification requirements. Of particular interest was the issues effecting the transient behavior of the thermal profile of the motor/generator, Figure 3-14. Available transient power surge is desirable to maintain an attractive power profile for short term acceleration and regenerative braking.

The electronics associated with the motor/generator were equally challenging. Conventional sensors for brushless dc motors were too slow. Therefore, an optional system was designed for the EDU testing. The associated commutation electronics also

incorporated some unique high speed circuitry. The motor drivers utilized the latest IGBT's. High speed FETS would have been preferred for the EDU but were not available in the required power rating (they were in development at that time and scheduled for production in one year, i.e. 1995-1996). The commutation system was developed using a twelve pole Durham motor running at lower speed. The EDU motor has fewer poles, is three phase, brushless dc, and is an ironless armature design.

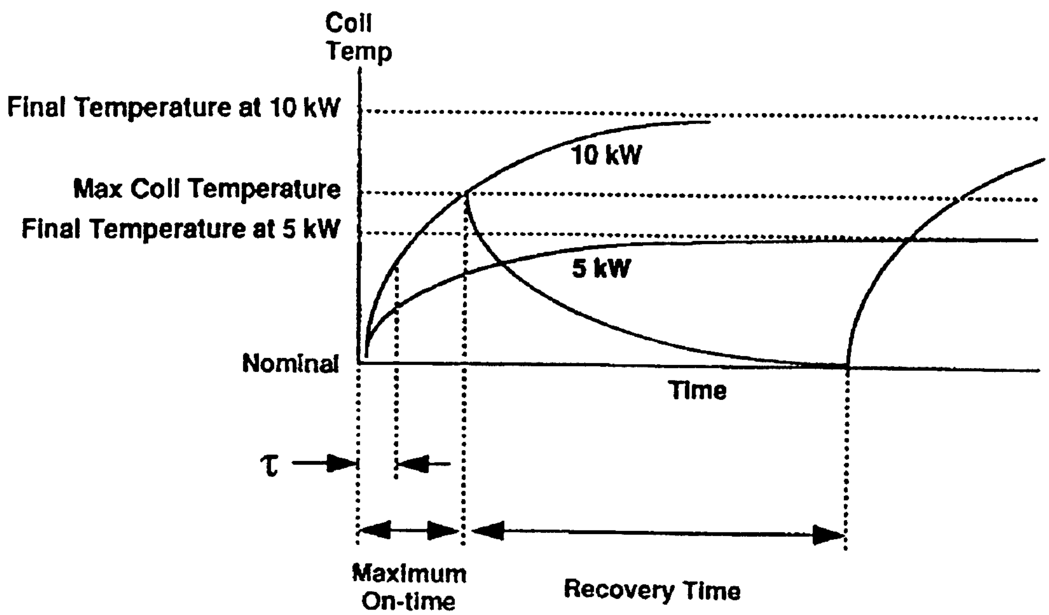


Figure 3-14 Motor / Generator Thermal Transient Power

Motor generator testing was planned in three phases. The first was to check out the commutation electronics and motor drives of the Durham multiple pole motor. This was accomplished using breadboard electronics and a simple vacuum housing (bell jar). This provided confidence in the optical instrumentation package and the commutation approach. Harmonics were reduced by minor changes to the electronics package. The next phase of testing was to utilize the breadboard electronics to run a prototype of the EDU motor/generator. Test results were favorable. Development then proceeded to the third phase; installing the motor/generator into the EDU test facility and testing in a vacuum at high speeds. Testing produced very satisfactory and encouraging results until a resonance was encountered at 63,000 rpm. The structure which supported the motor stator was the problem as a structural resonance condition terminated the testing. Due to Honeywell's cessation of efforts on this program, no further testing of the HSSO motor/generator design was initiated for AFS.

3.1.4.3 Results Summary / Key Accomplishments

There were many achievements and intermediate milestones attained by the management and technical staff on this portion of the American Flywheel System program. Many of these have been detailed in the foregoing paragraphs. Certainly the application of new analytic techniques coupled with innovative designs offered new approaches to component development. Unfortunately to date only the motor / generator has been tested. Other hardware designs from HSSO, as well as technology from ORNL and UVa, await the opportunity to demonstrate their capabilities in the EDU test facility. The EDU test fixture became operational at Honeywell in time to provide an adequate capability to evaluate the motor / generator to 63,000 rpm. The outgassing and pin-on-disk tribology testing proved valuable for material selection and confirmation. Delivery of a functional Demonstration System to AFS early in the program and completion of the EDU optimized design rank high among those accomplishments.

The list of what was accomplished, just on the Honeywell portion of the AFS program, is a lengthy one. Therefore to summarize, Table 3-6 presents a tabular form of the milestones and accomplishments in the Honeywell portion of the AFS program for the years of 1993 and 1994. Honeywell's final monthly report to AFS covered the period to September 1994.

Table 3-6 EDU Program Accomplishments/Milestones

Date	1993 EDU Program Accomplishments and /or Milestones
May	**Program kickoff meeting with AFS** *(Review of detailed workplan as submitted by Honeywell for AFS approval.)*
August	**Demonstration system delivered to American Flywheel System.** *(This unit provides insight into the operation and principals of the EMFB. It use is educational as well as promotional in elevating the understanding of mechanical battery operation.)*
September	**Technical due diligence completed.** *(Make or buy decisions on key EDU components by program team. Agreement to subcontract composite rotor design / manufacture.)*
September	***System engineering plan completed. Strawman EDU design completed.*** *(Honeywell completed outline and detailed development of system engineering plan for the development of the EDU and prototype EMFB.)*
September	**Material survey for magnetic bearing application and for pin-on-disk testing.** *(Key material selection process commences with tribology selection process for wear / friction determination completed.)*
October	**Review of composite rotor proposal completed. Subcontractor selected.** *(Oak Ridge National Laboratory selected as vendor of choice for composite rotor.)*

November	**Honeywell Re-plan submitted to AFS for approval.** *(Honeywell identifies limitations and technical challenges which require restructure of the EDU portion of the AFS contracted effort, for progressive component "build-up" and testing to complete.)*
December	**Preliminary design of EDU test rig.** *(First design of EDU test fixture based upon the recommended re-plan is completed.)*

Date	1994 EDU Program Accomplishments and /or Milestones
January	**Optimization analysis and software operational.** *(Numerous EMFB / EDU designs evaluated ranging from 2 kWhr to 14 kWhr for electric vehicle deployment.)*
February	**Five (5) patent disclosures transmitted to AFS** *(Patents disclosures are related to specific system related activities as well as the rotor itself:* *1. Composite flywheel rim and hub interface* *2. Gimbal mount for flywheel* *3. "Ganged" gimbal system for flywheels* *4. Burst containment vessel* *5. Energy absorbing housing.)*
March	**One (1) patent disclosure transmitted to AFS.** *(Patent disclosure related to the magnetic bearing design as selected by HSSO.)*
April	**Re-plan transmitted to SMUD for approval by AFS.** *(After review of additional AFS requested justification of the replan, AFS request of SMUD that the program continue with the Honeywell EDU activity as modified, saving judgment on the prototype portion of the re-plan.)*
April	**One (1) patent disclosure transmitted to AFS .** *(Patent disclosure related to the motor / generator design as selected by Honeywell Durham and Phoenix operations.)*
April	**3.6 kWhr / 8 kW EMFB design selected by AFS.** *(After final tuning of the optimization analysis and approach, HSSO offers a revised design for AFS consideration.)*
June	**EDU test plan developed.** *(Full test plan outlining test instrumentation and procedures is submitted to AFS for approval.)*
June	**Four (4) patent disclosures transmitted to AFS.** *(Patents disclosures are related to system components under development;* *1. Power converter* *2. Magnetic bearing* *3. Motor/generator* *4. Vacuum seal.)*

July	**EDU Structural dynamics report transmitted to AFS.** *(Report is first compete systems dynamics analysis of rotor assembly including finite element evaluation of structure and rotor dynamics performance.)*
August	**Outgassing materials test report transmitted to AFS.** *(Outgassing by Lockheed provided direction for continuation of composite structure and rotor development.)*
August	**Draft patent disclosure transmitted to AFS** *(Patent disclosure is on modified and updated EMFB design.)*
August	**Tribology test report (pin-on-disk) transmitted to AFS.** *(Touchdown bearing mechanism and materials selected.)*
August	**EDU test fixture design and fabrication completed.** *(Test fixture now available to commence high speed vacuum testing of components.)*
September	**Motor / generator tested to 63,000 rpm at HSSO.** *(Structural dynamics of stator fixturing proves dynamically sensitive and requires revision.)*
September	**EDU design review with Oak Ridge National Laboratory personnel.**

3.1.5 Overview of ORNL Effort

3.1.5.1 Introduction to ORNL Effort

The following sections present work performed at ORNL under the funding and direction of AFS to develop a high performance composite rotor to efficiently store energy for the EDU. The composite rotor is a critical component of the AFS Electro-Mechanical Flywheel Battery (EMFB). The rotor stores the kinetic energy that is converted to electrical power by rotating at a very high rate of speed. The faster the rotational speed, the greater the energy stored, resulting in higher performance (watt-hours per pound) of the flywheel battery. The rotor/hub design has been developed at the ORNL Composite Manufacturing Technology Center for AFS. The design is owned by AFS and utilizes the ORNL technology base developed over the past 30 years which has established worlds records for flywheel composite rotor specific energy (watt-hours/pound).

Objectives of the Composite Rotor Development Program:

The objective of this program was to produce an Engineering Development Unit (EDU) rotor to demonstrate maximum capability of specific energy. Potential production issues and areas of improvement that would enhance the performance and producibility of the composite rotor (lessons learned) were addressed. ORNL's goal was to strive for a rotor design that would achieve a significant improvement in the operating speed of the rim and the maximum specific energy.

Technical Issues:

Suppliers of continuous filament reinforcement materials have developed high performance fibers with strengths approaching and sometimes exceeding 1,000,000 psi. However, resulting composite performance is strongly process-dependent. In addition to process variations the continued presence of significant property variations within each successive production run of fiber was, and still is, a concern. Key mechanical properties of composite specimens and components made with specific lots of reinforcement fibers were characterized in order to assure proper material selection and design parameters. Detailed stress/strain analysis, critical speed analysis, stability analysis, and response analysis were conducted to validate the performance of the complete rotor assembly.

Scope of Work (EDU):

Task 1- Material Characterization includes;

1) Characterization and then selection of the best high-performance fiber,
2) Strand Ultimate Tensile tests of selected fiber,
3) Strand Stress Rupture testing,
4) NOL Split-D Ring Ultimate Strength testing,
5) NOL Split-D Ring Cyclic Fatigue testing.

Task 2- EDU Design;

1) Develop Optimum Rotor Design including design trades and feasibility assessments,
2) Perform detailed Stress/Strain analysis,
3) Perform Dynamic Analysis, Critical Speed Analysis, Stability Analysis, and Response Analysis,
4) Prepare Drawing Package.

Task 3- Fabricate Rotor.

Task 4- EDU Rotor Test Design;

1) Develop Rotor Test Plan,
2) Develop Design for EDU Spin Test Fixture.

3.1.5.2 Technical Challenges / Approach

The following list summarizes the flywheel rotor design challenges faced by AFS and their subcontractor ORNL.

Flywheel Design Challenges:

- Design Trades and Optimization
    - Energy and Volumetric Efficiency
    - Radial Stress Control
- Manufacturing Considerations
    - Producibility
- Safety and Composite Rotor Performance
    - Safety
    - Low Outgassing
    - Compatible With Operating Environment

- Maximize Composite Ultimate Strength
- High Creep / Stress Rupture Allowable
- Minimize Fatigue Sensitivity-Low Cycle Fatigue (LCF), High Cycle Fatigue (HCF)

- Rotor Performance Verification

Design Trades and Optimization:

Two critical issues were addressed to achieve maximum performance. The first was the optimization tradeoff between specific energy (*Wh/lb*) and energy density (*kWh/m³*). Figure 3.1.5.2-1 shows the typical energy and volume trade. A thinner rotor stores more energy per weight of rotor. However, a thinner rotor takes up a larger volume which can limit the number of flywheel batteries that can be placed in a car. This trade is critical for the automobile application.

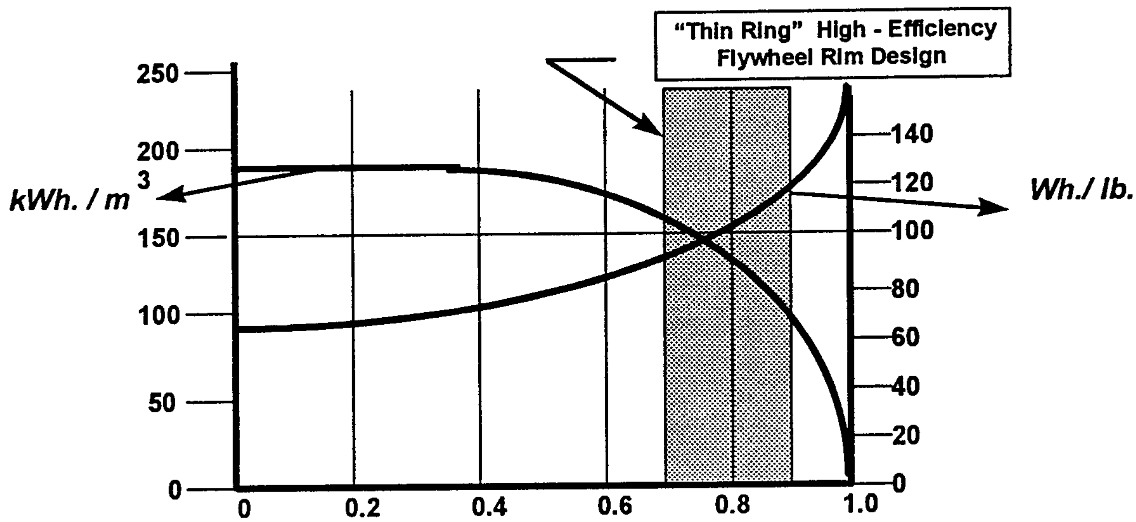


Figure 3.1.5.2-1 Energy and Volumetric Trades

The second critical issue was the method of handling radial stresses in the rotor. When a rotor is optimized for maximum tangential stress, the radial stress becomes a critical design consideration for rotors with rim thickness above approximately 0.5 inch and also at the interface between the rotor and the hub. There are two preferred approaches to handling radial stresses: nested rings that are pressed together with an interference fit or a multiple material rim where the inner materials loads the outer materials thus minimizing radial stresses. This analysis utilized the NEST Program which is a special purpose nested ring analysis program.

Manufacturing Considerations:

Because of the high strength requirements, the rotor was fabricated utilizing a wet filament winding process to achieve a high fiber volume, low void content, and high strength composite (Table 3.1.5.2-1). This AFS design, consisting of an all graphite/epoxy rotor/hub assembly, provides maximum specific energy by: first

maximizing the mass in the rim and minimizing the mass in the hub, second using a filament winding process which produces the highest fiber volume composite resulting in the highest hoop strength rotor, and third assembling the rotor using interference fit to minimize radial stresses.

Table 3.1.5.2-1 Producibility-Composite Manufacturing Considerations

PARAMETER	WET FILAMENT WINDING	PRE-PREG WINDING	COMMENTS
FIBER FRACTION	78-80%	57-62%	Firm Data
REPRODUCABILITY	Yes	Yes	Mfg. Demonstrated
MATERIAL COST	Lower	Higher	Manufacturer Data
CAPITAL COST	Lower	Higher	More Process Steps
PRODUCTION OUTPUT	High	High	Not Discriminator
VOIDS	Low	Low	Not Discriminator
TRANSVERSE PROP. -STRESS	Lower	Higher	Design Requirement Factor
TRANSVERSE PROP. -MODULUS	Higher	Lower	Design Requirement Factor

Safety and Composite Performance:

The composite rotor/hub performance was validated through extensive materials and component testing which included:

- Outgassing
- Strand ultimate tensile strength
- Strand stress rupture
- Creep
- Split-D ring (NOL) cyclic fatigue

Rotor material testing results indicate that this flywheel rotor will achieve a new specific energy record for a practical flywheel rotor configuration. Test results have also demonstrated a composite design allowable consistent with an operational life of 30 years with 11,000 discharge cycles. The resulting composite rim has a fiber volume of 78% with a void content less than 1%. Outgassing tests have determined that this composite rotor will also meet NASA and Military outgassing specifications.

Rotor Performance Verification:

Detailed stress/strain analysis, critical speed analysis, stability analysis, and response analysis has been performed to validate the performance of the complete rotor assembly. Table 3.5.1.2-2 presents analyses that were performed to demonstrate rotor assembly performance. ABAQUS and NISA are both 3-D Finite Element Analysis (FEA) programs and were used to determine stresses, strains, modes and mode shapes. Finite element analysis using a model with 100,000 Degrees of Freedom (DOF) verified positive stress margins, acceptable strain levels, and determined the optimum interference fits for minimizing radial stresses in the composite epoxy.

Table 3.1.5.2-2 Analysis Models Demonstrate Rotor Assembly Static and Dynamic Performance

MODEL	ABAQUS 3-D FEA	NISA 3-D FEA	FEATURE BEAM	ANALYTICAL SOLUTION
ROTOR	X	X	X	
RIM	X	X	X	
HUB	X	X	X	
SPRING-MASS	X	X		X

The rotor design was also optimized to move all flexible modes as high as possible, preferably out of the operating speed range of the rotor. The Finite Element Analysis Tool for Utility Rotor Evaluation (FEATURE), a unique rotor - bearing dynamics program which calculates critical speeds, stability, and response of high speed rotating equipment, was utilized to optimize and verify the high speed performance of the integrated rotor/shaft/bearing assembly, including the effects of bearing stiffness and damping. The following summarizes the information obtained from each type of FEATURE analysis:

<u>Critical Speed</u>

- Non-rotating natural frequencies and mode shapes
- Synchronous critical frequencies and mode shapes
- Subsysnchronous critical frequencies and mode shapes
- Supersynchronous critical frequencies and mode shapes

<u>Stability</u>

- Damped asynchronous whirl frequencies and mode shapes
- Log decrements for damped modes
- Stability determination of sign of Log decrement
- Regressive vs. progressive whirl identification
- Whirl orbit

<u>Response</u>

- Forced synchronous response vs. frequency
- imbalance response
- Forced asynchronous response
- Response orbit

3.1.5.3 Results Summary and Key Accomplishments

<u>Rotor Preliminary Design</u>:

The goal of achieving the highest performance rotor in the smallest package at the lowest cost started with the material selection. Many materials were considered based on vendor data with two being selected for performance verification testing. The driving requirement was Ultimate Tensile Strength (UTS), but Modulus was also a consideration. Both materials evaluated (see Table 3.1.5.3-1) met the requirements for Modulus but

Toray 1000G had higher UTS and also was half the cost, resulting in a specific performance (UTS/$) 2.25 times the alternate material.

Table 3.1.5.3-1 Material Selection Process-Relative Material Performance

PROPERTIES	TORAY T1000G	ALTERNATE MATERIAL
Strength (UTS)	*100%	*89%
Standard Deviation	Lower	Higher
Coefficient of Variation	Lower	Higher
Specimens Tested	143	342
Modulus	*100%	*104%
Standard Deviation	Lower	Higher
Coefficient of Variation	Lower	Higher
Specimens Tested	29	72
Cost	100%	200%

***Toray properties normalized to 100%**

The next step in the rotor design process was to evaluate rotor designs given the requirements for total energy stored and maximum operating speed. Because both radial stress and tangential stress are critical, a special purpose "Nested Ring Analysis Program" (NEST) is required to develop an optimum rotor design. Table 3.1.5.3-2 describes the features of the NEST program.

Table 3.1.5.3-2 Special Purpose Nested Ring Analysis Program Description

• Performs stress analysis of assemblage of nested orthotropic thick rings.	
• Formulation -	Axisymmetric, plane stress elasticity solution with interfacial discontinuity conditions.
• Applied Loads -	Centrifugal loads due to constant angular velocity. Internal and/or external radial pressure. Uniform temperature change. Electromagnetic loads (simulated via. material density and rotation. Initial mechanical interference between rings.
• Capabilities -	Ring thickness can be varied (discrete step or continuous taper).
• References -	S.G. Lekhnitskii, "Anisotropic Plates", Gordon & Breach; Toland & Alper, Transfer Matrix for Analysis of Composite Flywheels", Journal of Composite Materials, Vol. 10, July, 1976

Three rotor rim configurations were evaluated; one material-one ring, two materials-one ring, one material-two rings (with interference fit between the two rings). The analysis varied rim thickness and length at the required constant speed and stored energy. Table 3.1.5.3-3 presents results from the two material-one ring analysis. As rim thickness increased, the specific energy (Wh/kg) decreases, as expected. Also, the S-Glass thickness increases with rim thickness which applied an increasing radial load on the graphite hoop wound material which controlled the radial stress. However, this S-Glass loading caused a larger increase in Hoop Stress as the rim thickness is increased. This analysis resulted in the final rotor rim design that was then evaluated with further analysis (detailed stress/strain analysis, critical speed analysis, stability analysis, and response analysis).

Table 3.1.5.3-3 Composite Rotor Design Selection Process; Typical Analysis- One Rotor Ring With Two Materials

Rim Thickness (in.)	S-Glass Thickness (in.)	Specific Energy (Wh/kg)	Radial Stress (psi)	Hoop Stress (ksi)
0.5	0.010	259.7	1030	490.7
0.6	0.027	255.2	1066	499.6
0.7	0.051	250.7	1055	509.8
0.8	0.080	246.3	1048	519.8
0.9	0.115	241.9	1031	530.0
1.0	0.155	237.6	1020	539.9
1.1	0.200	233.3	1014	549.4
1.2	0.250	229.1	1010	558.6
1.3	0.305	225.0	1007	567.3
1.4	0.360	221.0	1066	573.6
1.5	0.425	217.0	1058	581.3

Figure 3.1.5.3-1 shows the dramatic increase in radial stress as the rim thickness increases (Ri/Ro decreases) for a one material-one ring. By using one material and two nested rings which are pressed together with an interference fit, these radial stresses can be reduced dramatically. At zero speed. The two rings are in radial compression, and at full speed the rings are near zero radial stress.

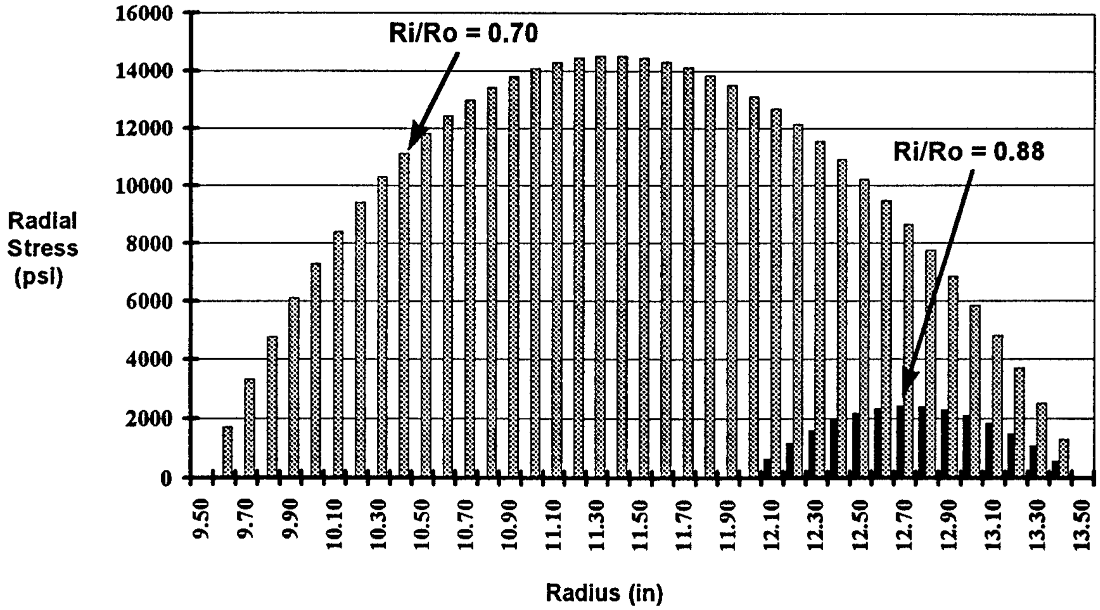


Figure 3.1.5.3-1 Example of Rim Radius Ratio (Ri/Ro) Effects on Radial Stress

Material Testing and Design Allowables

Upon selection of Toray 1000G fiber, addition fiber was ordered (two lots) for additional testing and rotor fabrication. The results of UTS testing summarized in Table 3.1.5.3-4 show that the fiber tows met the expected strength of 900 ksi. Some samples were as high as 940 ksi.

Table 3.1.5.3-4 TORAY 1000G Strand Tensile Tests

	Lot 1	Lot 2
Strength (UTS)	916.6 KSI	887.6 KSI
Standard Deviation	49.6 KSI	49.9 KSI
Coefficient of Variation	5.4 %	5.6 %
Specimens/Spools	241 / 16	99 / 8
Modulus	40.0 KSI	40.2 KSI
Standard Deviation	1.1 MSI	1.2 MSI
Coefficient of Variation	2.8 %	3.1 %
Specimens/Spools	48 /16	24 /8

Since rotor operation requires that the rotor be at speed for most of its operating life, it is essential that this requirement be considered when setting the operating design stress. Composite characteristic life versus percent characteristic strength as interpreted by the "Two-Parameter Weibull Distribution" must be developed. The best method for developing this allowable is through the use of "Strand Stress Rupture" data. Strands are hung at five stress levels (see Table 3.1.5.3-5) until they break in order to develop a

database. Typical strand stress rupture results at 83.5 percent of ultimate are presented in Table 3.1.5.3-5.

Table 3.1.5.3-5 Strand Stress Rupture Test Description & Test Examples

- **200 specimens**
  - **50 - Ultimate Tensile Strength**
  - **150 - Stress Rupture**
- **5 Stress Levels**
  - **97%, 93%, 89%, 83.5%, and 80%**
  - **30 specimens per load condition**

Specimen	Percent of Ultimate (%)	Time To Failure (Sec)	Comment
2	83.5	12,068	
9	83.5	754,964	
16	83.5	0	Broke during loading
29	83.5	3,463,091	

The Two Parameter Weibull distribution is an approach to calculating the life versus %UTS that has substantial empirical support with models accurately fitting the data. Table 3.1.5.3-6 summarizes the theory behind the Two-Parameter Weibull distribution and also presents the properties of the Weibull distribution. Figure 3.1.5.3-2 shows a typical Life/Stress Rupture plot with 10 year and 30 year life lines indicated. Using B0.15 (0.15% failure) and 10 year life results in a design allowable of ~82% UTS. At 30 years the design allowable is ~80% UTS. This approach was used to develop the design allowable for the AFS rotor as shown in Table 3.1.5-7.

Table 3.1.5.3-6 Design Allowable Determination; Weibull Statistics

The two-parameter Weibull distribution is preferred by MIL-HDBK-17-1C (and ORNL) based upon the following factors:

-Theory suggest the Weibull is appropriate for strength distributions of brittle materials such as fibers,

-"Chain-of-Bundles" model for unidirectional composites suggests a Weibull model as appropriate for strength distributions,

-Empirical support is provided by models fit to data,

-Lifetime distributions have been shown to be well modeled by Weibull.

<u>Properties of the Weibull distribution</u>

n (characteristic strength, life)
-load on time at which 63.2% of units fail

b (shape factor)
-measure of variability
b< 1.0 implies infant mortality
b= 1.0 implies random failure
1.0< b< 4.0 implies early wear out
b> 4.0 implies old age wear out

B (life notation)
B10 (10% failure) - MIL-HDBK-17-1C B Basis
B1 (1% failure) - benign failures (A Basis)
B0.15 (0.15% failure) - ~"3 sigma margin"
B0.10 (0.10% failure) - serious failures
B0.01 (0.01% failure) - catastrophic failures

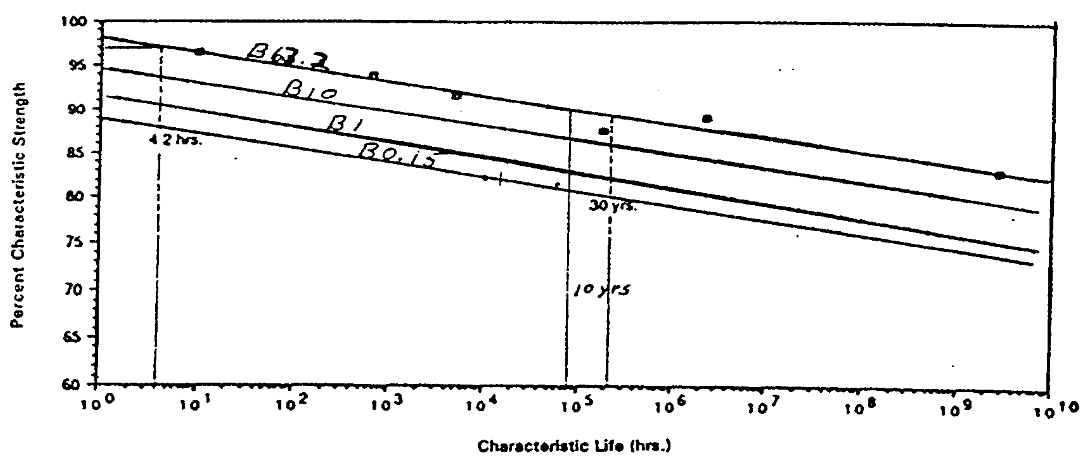


Figure 3.1.5.3-2 Typical Life/Stress Rupture Plot

Table 3.1.5.3-7 Typical Design Allowable Calculation; ORNL Stress Life Basis

Typical Strength of Fiber		900000 psi
Shape Factors	25.69	
No. Samples	113	
Characteristic Strength of NOL		628480 psi
Shape Factor	35.21	
No. Samples	24	
Fiber Fraction		0.783
Apparent Fiber Strength in NOL psi		802657
"Translation" Factor		0.89
Volume Reduction Factor - NOL > RIM		0.989
RIM Strength		621567 psi
DESIGN ALLOWABLES		
(10 years @ 0.15 % failure)		
0.50 beta = 0.8135		505645 psi
0.25 beta = 0.7363		457660 psi
(10 years @ 1% failure)		
0.50 beta = 0.8376		520624 psi
0.25 beta = 0.7805		485134 psi

NOL Split-D Ring composite testing represents one key element of the composites test program leading to the characterization of a high performance, safe composite rotor. Split-D ring testing simulates the hoop loading condition on a spinning rotor and verifies the analytically predicted performance of the rotor. Split-D testing is the most accurate method to determine the ultimate hoop strength and stiffness of the composite rings under hoop tension loading (Table 3.1.5.3-8). The composite test specimen consisted of a 24 inch diameter ring that was 0.125 inch thick and 0.25 inch wide. The Split-D loading fixture was attached to a typical tension test machine which records load and deflection from which the hoop strength and stiffness were determined. The advantage of this Split-D test fixture is that the test specimen is loaded in its actual structural configuration, in as accurate a simulation of the actual hoop tension loading condition as possible. Results of the testing have demonstrated hoop strengths which meet the design goal for the EDU. This test fixture was also used to demonstrate the required cyclic fatigue life of the composite rotor and to demonstrate the safe operating speed. Typical cyclic fatigue are presented in Table 3.1.5.3-9. The Split-D test rings were manufactured using a wet filament winding process where dry tows of graphite fibers were wetted in a resin bath and

then wound on a mandrel. This process resulted in a very high quality, high fiber content composite which is critical to the high performance of the rotor. After the winding was completed and the cylinder cured, individual composite test rings were cut from the composite cylinder for testing. This wet filament winding process is identical to the manufacturing process for the flywheel rotor where longer thick walled cylinders are wound and cured and then cut to the desired length resulting in multiple rotors from one filament wound part.

Table 3.1.5.3-8 NOL Ring Testing

Specimens Fabricated -Toray T 1000G
- 76 % Fiber Volume
- 24 inch diameter
- 1/8 inch wall thickness x ¼ inch wide
Approximately 60 Rings
- 21 Tested for Static Strength
- 20 Cyclic Fatigue
- 19 Spares
Four Mean Stress Levels
- 90 %, 85 %, 80 %, and 75 % UTS
- 5 Rings per Load Condition
NOL Ring Test Machine / Fixtures
- MTS Servo-hydraulic Test Machine
- 200 KIP
Test Conditions
- Room Condition
- Air Environment
- 0.5 Hz
- Cycle From 500 lbs. To Maximum Load

Table 3.1.5.3-9 Example Cyclic Fatigue Results

Ring Number	Percent of Ultimate	Number of Cycles**	Comments
8	79	1540	
14	80	5447	
31	80	4574	*

* Broken Stud-Ring , NOL Did Not Fail

** Cycles To Failure

Figure 3.1.5.3-3 presents one of the many detailed stress analyses that were performed during the design of the rotor assembly. This NISA analysis was used to design all critical components of the rotor. Stress concentrations were analyzed in detail to optimize the design and to minimize stress concentrations as shown in Figure 3.1.5.3-3.

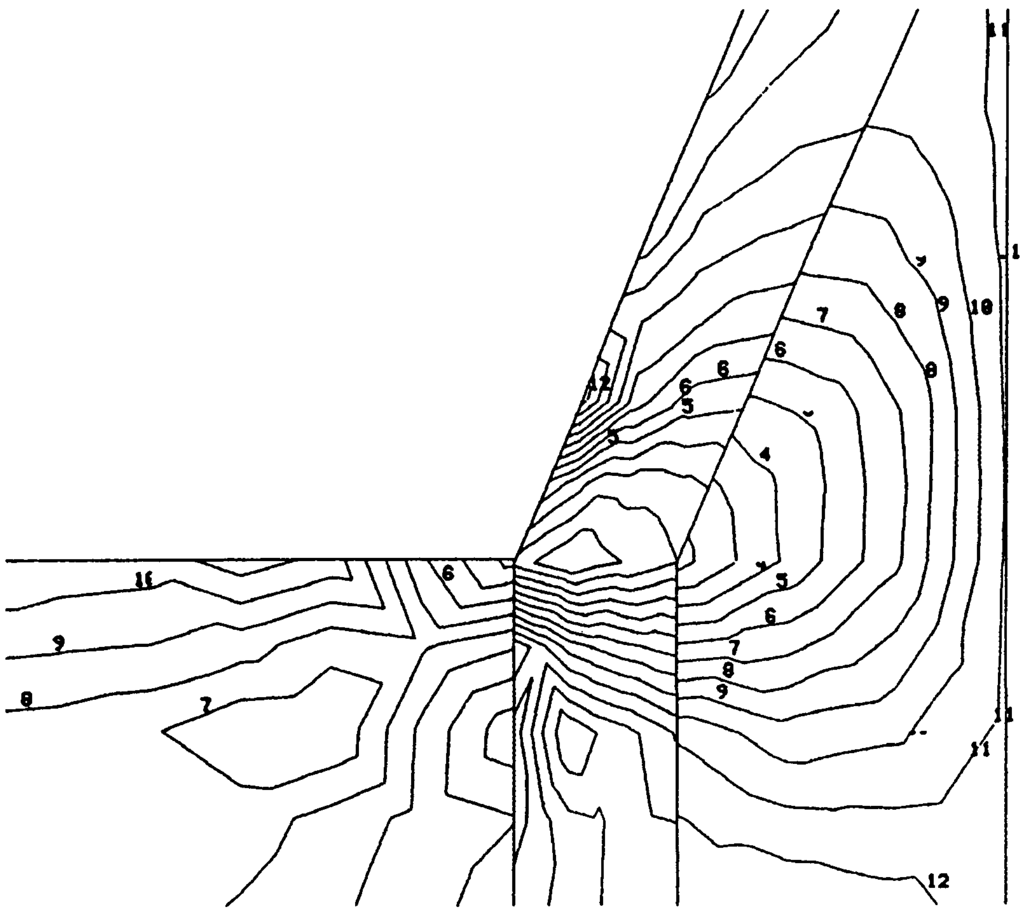


Figure 3.1.5.3-3 Example Detailed Stress Plot

Dynamic, Critical Speed, Stability and Response Analysis:

The necessity of offering a stable platform for the operation of the high speed EDU is mandated by both safety concerns and system performance issues. The interaction of components in the rotor-dynamics evaluation is generally found to be extensive. Therefore, the assessment of rotor-dynamic performance must include all key components of the system such as the rotor, the bearings, the bearing support structure, the drive turbine etc. At high speeds all rotor systems may encounter some degree of flexible behavior. To deal with the issue of system dynamics in the design phase, good predictive tools are essential. Software resources were made available to ORNL as developed under funding from the Electric Power Research Institute (EPRI) and managed by Mechanical Technology Inc. (MTI). The code implemented is called FEATURE and it is a finite element based resource which included the capability of predicting, among other things;

- Natural Frequencies
- Critical Speeds
- Stability
- Imbalance Response
- Response to a-synchronous loads and excitation

Per Figure 3.1.5.3-4 as the rotor spins up in speed it encounters and traverses (on the diagonal line) a number of natural frequencies (whirl or processional frequencies as noted). The dominate precession (whirl) motion of the rotor may be in the direction of spin (forward) or in a direction opposite to spin (retrograde). As the rotor encounters a forward precession natural frequency (mode) it is called a "critical speed". Table 3.1.5.3-10 lists in order of increasing frequency the critical speeds of the spinning assembly. The term critical speed is appropriate as the rotor tends to execute large motions (orbits) as it traverses this point and the state of balance of the rotor is indeed "critical" to its success. As illustrated per Figure 3.1.5.3-5 the FEATURE analysis provided a multitude of frequencies of interest within and outside the dynamic range of the system. The illustration presents the exaggerated deformation of the rotor and components when encountering this natural frequency (mode). Lateral motion in mode 1 provides that the rotor deforms such that the ends of the shaft are in phase whereas mode 2 is angular indication the ends of the shaft are out of phase. The location of natural frequencies/critical speeds is generally controlled by the bearing stiffness properties and the stiffness and mass of the rotating assembly. As the dynamics of the EMFB became more refined through better understanding of the bearing suspension and rotor design, ORNL recommended rotor tuning and adjustments which were accommodated by AFS to secure the best design for further consideration.

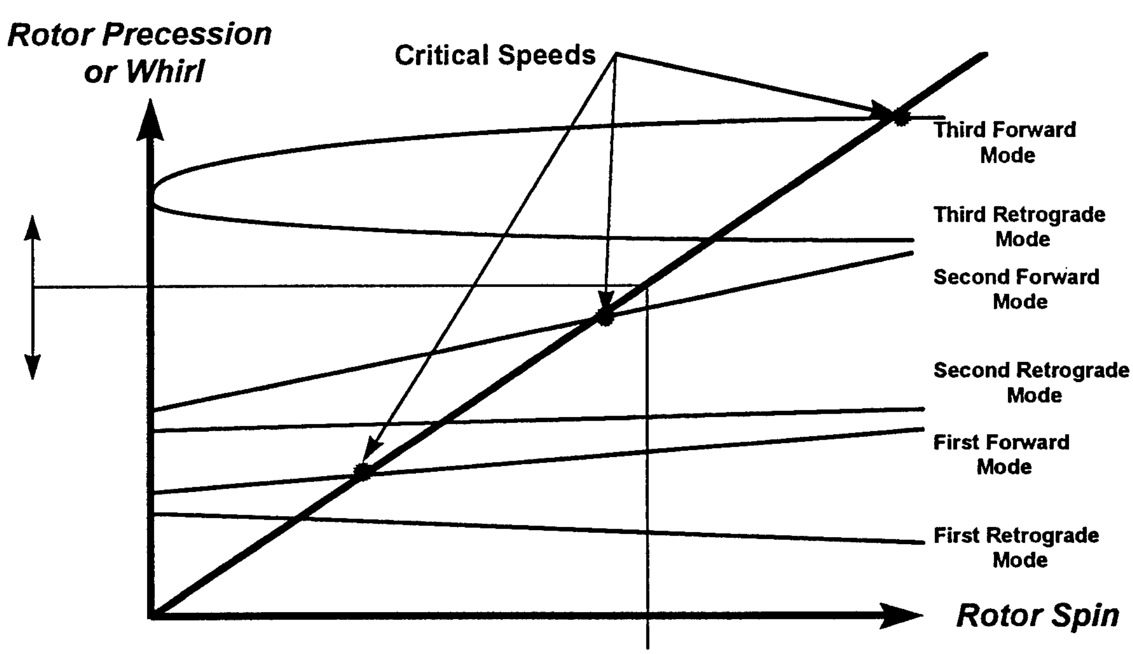


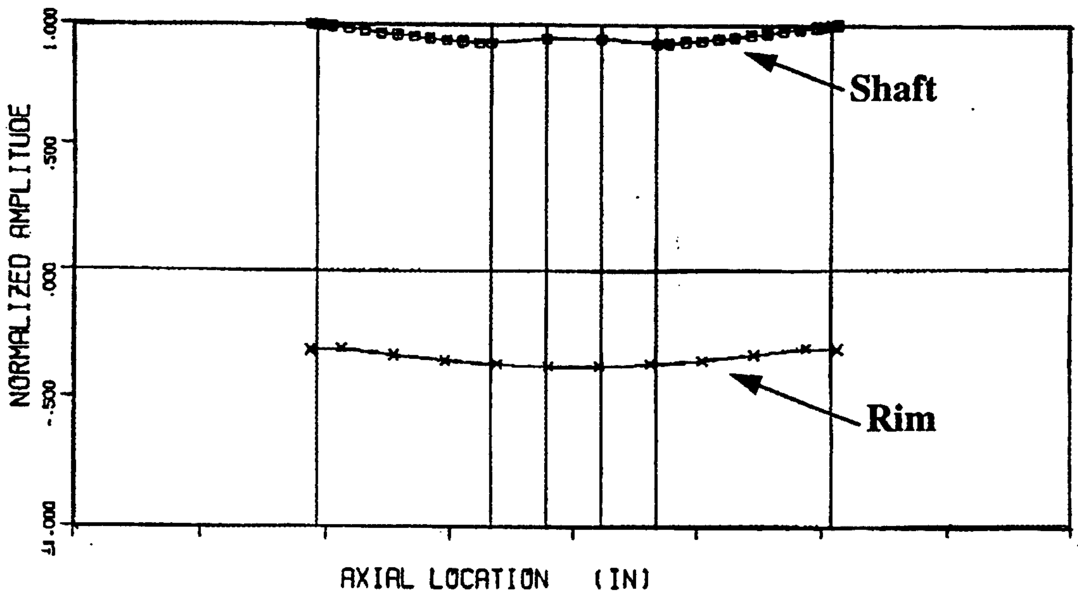
Figure 3.1.5.3-4 Dynamic Analysis

In the past, many rotor systems were designed with an understanding of critical speed and imbalance control, yet still encountered severe dynamic behavioral problems. In the 1970's, the comprehension and prediction of stability behavior of high speed rotors became available to the designer. Stability assessment requires understanding of the modes of vibration of the system, even though they are not being excited by imbalance of the rotor. The natural frequencies of vibration can be excited by any available perturbation such as from; road bumps, rail ties, pot holes and alike. As the rotor is spinning at high speeds, the excitation of an asynchronous natural frequency (i.e. not a critical speed) is not generally controllable by balancing. This issue is common to many high speed rotors such as gas turbines, auxiliary power units, turbine feed pumps, LST-G etc. The most effective remedy is to insure that the rotor can be controlled by the bearings (tuning of mode shapes) and that the bearings provide adequate damping to dissipate the unwanted energy and prevent the system from becoming unstable and self destroy. As illustrated per Table 3.1.5.3-11, the logarithmic decrement, which is used as a measure of stability (stable if number is positive), is indicating stable performance for the ORNL design.

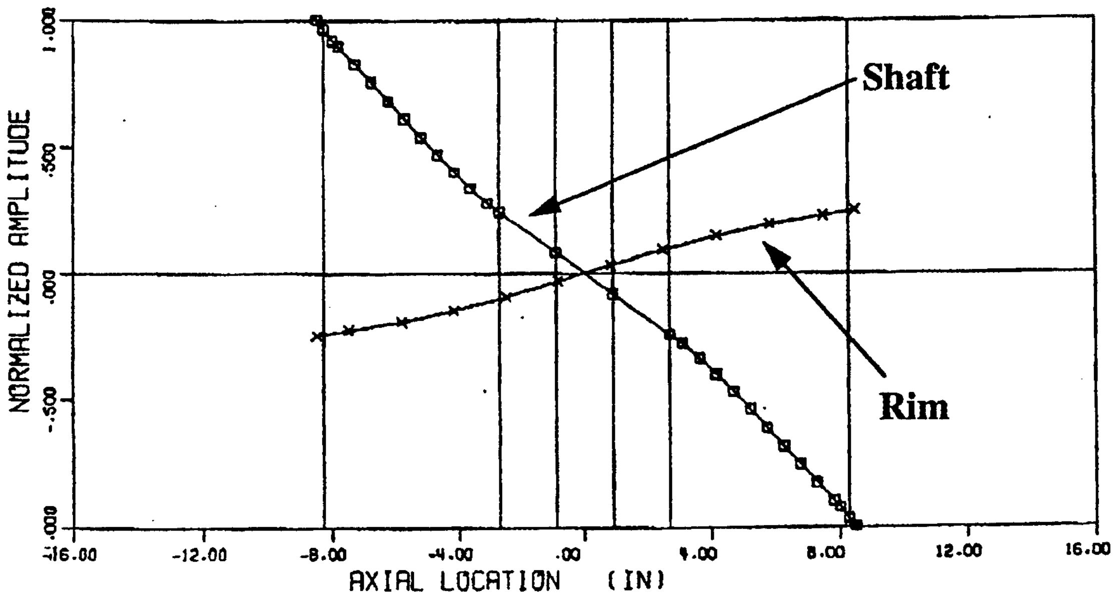
Imbalance response is not only important at critical speeds but for any operating condition. Good balance equipment and practices minimize imbalance forces and are required for the EDU rotor as well. Forced (or imbalance) response is as illustrated per Table 3.1.5.3-11. Values of bearing loads reported are very sensitive to the distribution and magnitude of imbalance conditions assumed present on the rotor.

Table 3.1.5.3-10 Typical FEATURE Rotor Critical Modes Analysis

Mode	Mode Identity
1	Lateral Rigid Body
2	Angular Rigid Body
3	Lateral
4	Angular
5	1st Shaft Flexural
6	1st Rim Flexural
7	2nd Shaft Flexural
8	2nd Rim Flexural
9	1st Hub Flexural
10	1st Bearing Pedestal Resonance
11	2nd Bearing Pedestal Resonance
12	3rd Rim Flexural



LATERAL (MODE 1)



ANGULAR (MODE 2)

Figure 3.1.5.3-5 Typical FEATURE Critical Speed Modes

Table 3.1.5.3-11 Typical FEATURE Stability and Forced Response Analysis

STABILITY ANALYSIS

Mode	Log Decrement	Quality Factor
Lateral (Mode 1)	0.0623	48.2
Angular (Mode 2)	0.1693	18.6

FORCED RESPONSE ANALYSIS

		Runout (mils S.A.)			Force (lbs)		
Inbalance	Mode	Pedestal	Bearing	Rim	Inbalance*	Suspension	Bearing
1g Mass	Lateral (Mode 1)	9.0	9.0	3.8	224 lbs	65 lbs	50 lbs
1g Couple	Angular (Mode 2)	18.1	16.9	4.7	636 lbs	187 lbs	187 lbs

*Force per one-half gram balanced weight (each end of rotor)

Component Fabrication:

All components for the assembly and test of two rotors were fabricated for AFS by ORNL using high performance composites as described in Table 3.1.5.3-12. A total of 18 parts were fabricated and are ready for assembly into two complete rotor assemblies.

Table 3.1.5.3-12 AFS Rotor Composite Components

Component	Parts	Material	Fabrication Method
Outer Rim	2	T100G/ERL2258-mPDA (78-80% VF)	Wet-Wind
Inner Rim	2	T100G/ERL2258-mPDA (78-80% VF)	Wet-Wind
Outer Band. Ring	2	T100G/ERL2258-mPDA (78-80% VF)	Wet-Wind
Inner Band. Ring	2	T100G/ERL2258-mPDA (78-80% VF)	Wet-Wind
Hub	2	T1000g (58-60% VF)	Prepreg Tow
Hub Banding	2	M46J/ERL2258-mPDA (69-75% VF)	Wet-Wind
Spacer Ring	4	T1000G/977-2 (64-66% VF)	Prepreg Tow
Growth Ring	4	S-Glass/ERL2258-mPDA (75-80% VF)	Wet-Wind

The rim, which is the most significant component of the rotor, is fabricated using a wet filament winding process which can rapidly lay down the desired thickness of composite material. Filament winding was selected not only for its performance, but also because of its adaptability to mass production. Although only one T1000G fiber tow was wound on this rotor rim shown in Figure 3.1.5.3-6, up to 20 fiber tows can be wound simultaneously for a production rotor rim. In addition, filament winding process variables can be controlled automatically, therefore requiring little manpower during the winding process. After removal of the rim from the mandrel, the rim is rough trimmed and press fit assembled with a second rim. Figure 3.1.5.3-6 shows the composite rotor rim after it is removed from the mandrel and before it is trimmed.

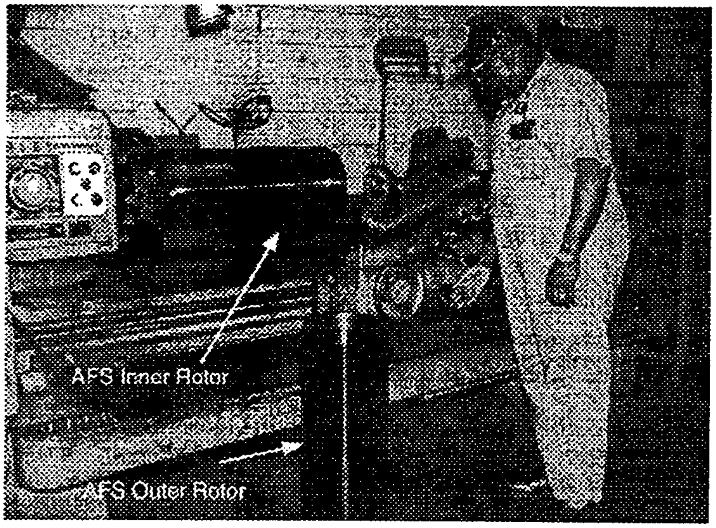


Figure 3.1.5.3-6 Photograph of the AFS inner and Outer Rims

Spin Test Fixture Design:

An EDU Spin Test Fixture was designed by ORNL (see Figure 3.1.5.3-7) and a set of preliminary drawings were developed. The Spin Test Fixture utilized the existing ORNL spin tank with the incorporation of a modified lid design that incorporates the EDU rotor spin support structure. This design also included the definition of multiple sets of instrumentation which were defined based on parameters to be measured, accuracy of measurement, and locations of measurement.

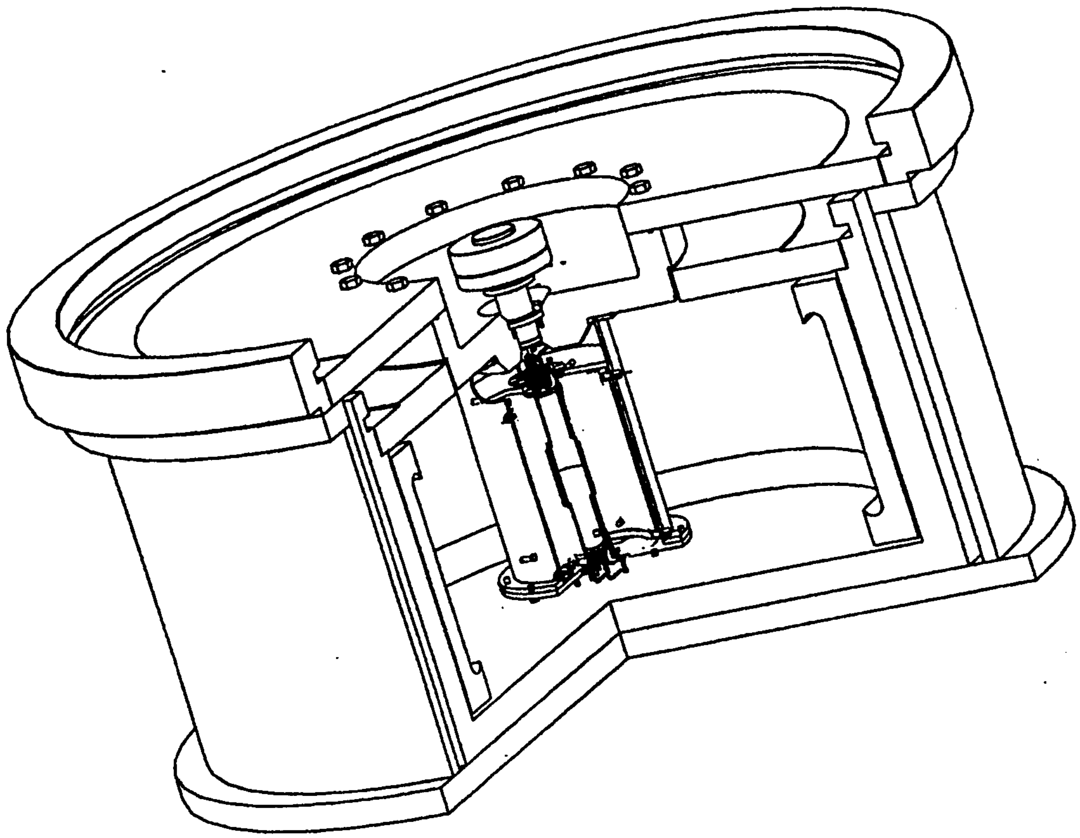


Figure 3.1.5.3-7 EDU Spin Test Fixture

3.1.6 Overview of University of Virginia (UVa) Effort

3.1.6.1 Introduction to UVa Effort

The University of Virginia (UVa) has Established the Rotating Machinery and Controls (ROMAC) Industrial Program. ROMAC currently supports about 50 industrial organizations in all elements of rotating machinery. Dr. Paul Allaire, UVa Center for Magnetic Bearings, has designed the magnetic bearings for AFS flywheel batteries on an exclusive basis. Dr. Allaire has previously demonstrated operation of magnetic bearings at 100K RPM for an avionics cooling compressor. The avionics cooling compressor test rig for the magnetic bearing is shown in the photograph in Figure 3.1.6-1. The high speed magnetic bearings are shown in Figure 3.1.6-2.

Figure 3.1.6-1: Photograph of Avionics Cooling Compressor Test Fixture

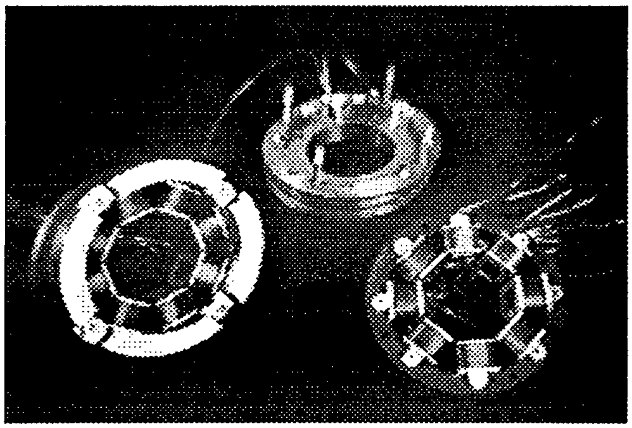


Figure 3.1.6-2: Avionics Cooling Compressor Magnetic Bearing Operated at 100k RPM

The AFS flywheel magnetic bearings designed by Dr. Allaire were developed as an alternative to the magnetic bearings designed by Honeywell. Because of the critical nature of this component in the EMFB, a comparison of alternative designs is well justified.

Objectives of the Magnetic Bearing Design Program:

- To offer an AFS alternative radial magnetic suspension system design consistent with maximizing specific energy density of the EMFB and without compromise to its intended deployment in an electric vehicle.

- To provide a preliminary design of that suspension with supportive technical information and documentation.

- To work closely on this effort with AFS and its other contractors during the completion of this effort.

Scope of Work:

Task 1.0 Review and Independent Assessment
Review all documentation and engineering detail as available and supplied by AFS regarding the axial and radial magnetic bearing specifications.

Task 2.0 Magnetic Bearing Alternative Recommendations
Determine a practical number of alternative approaches and configurations of magnetic bearings which could support the mission of the EMFB. Review those configurations with AFS and secure AFS' concurrence with the recommended alternative prior to proceeding.

Task 3.0 Magnetic Bearing Design of Suspension System
Upon concurrence from AFS, develop a design of an axial and radial magnetic suspension system providing AFS with all supportive analysis and a complete set of detailed drawings.

3.1.6.2 Technical Challenges / Approach

The energy storage flywheel is supported in magnetic bearings and associated electronic controls. The use of magnetic bearings are essential to high speed flywheel operation to provide a high DN (D= Diameter in mm. and N = RPM, in the range of 4 to 5 million) which is beyond the capability of rolling element mechanical bearings. Magnetic bearings have the needed capability to operate without lubrication at very high rotational speeds with low losses and at high (and low) temperatures. Magnetic bearings require an electronic control system which must be designed accounting for the particular physical rotor dynamics characteristics known for the flywheel rotor, the motor-generator properties and the magnetic bearing shaft.

Critical magnetic bearing design objectives for energy storage flywheel applications include as a minimum: compact low weight design, minimum power loss and high specific static and dynamic load capacity. These particular design criteria and the magnetic bearing design approach to achieve this program's objectives are discussed below.

Challenge 1: Compact/Low Weight Design

Both axial (thrust) and radial bearings are required. A double acting thrust bearing and associated thrust disk will be installed in the unit to take loads in either axial direction, regardless of shaft orientation. Two radial bearings and associated magnetic bearing shaft components, including rotor lamination stacks, will be required.

Compact actuators with low weight are essential to energy storage flywheels for automobile, satellite, and other applications. The flux paths, coil geometry, and rotor design must be very efficient. The University of Virginia (UVa) has designed, constructed, and delivered industrial magnetic bearing actuators for a high performance aircraft application operated to 100,000 rpm at 10 g. A photograph of these bearings was shown in Figure 3.1.6-2.

Challenge 2: Minimum Power Losses

Rotor losses are a key factor in magnetic bearing design for flywheel applications. Significant magnetic bearing losses will drain the useful power of the flywheel over a relatively short period of time. The NASA, Lewis Research Center, sponsored research project, including the world's most advanced experimental magnetic bearing loss test rig (see Figure 3.1.6-3) and extensive computational modeling at UVa for the past 5 years, has lead to much greater understanding of magnetic bearing loss phenomena.

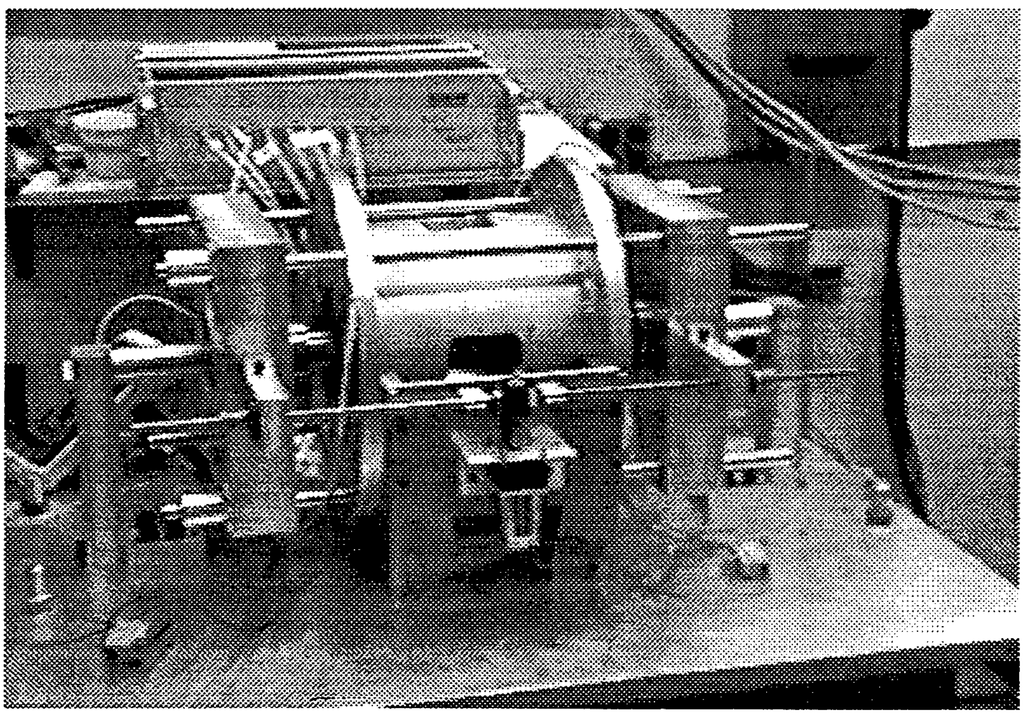


Figure 3.1.6-3 Magnetic Bearing Loss Test Rig at the University of Virginia

The dominant losses in a magnetic bearing dominant losses at these speeds are due to rotor eddy currents, and not due to the stator coil current losses, the rotor magnetic hysteresis losses, or the windage losses.

Challenge 3: High Specific Static and Dynamic Load Capacity

This application will require the development of high specific loads: the ratio of load to weight. High load capacity is a primary design factor. High force capability requires the operation of the bearings in the saturation range of the magnetic materials. A problem with conventional magnetic bearing current feedback control methods is that the actuator force is not well correlated to the current feedback loop for dynamic load capacity when in the saturation range. Effective use of magnetic bearings in the saturation range is best obtained with the use of flux feedback. A unique method for accomplishing this has been developed at UVa.

Finite element analysis and industrial experimental results indicate that magnetic bearings with large angular extent poles are subject to local saturation effects. These effects result in the loss of load capacity when high magnetomotive forces are applied to the bearing. The bearings supplied for this program will avoid this problem by employing extensive finite element modeling in the design stage.

Challenge 4: Magnetic Bearing Controller

Magnetic bearings require feedback control systems to operate. The critical design objectives for the control system are:

1. Stable Feedback Controller
2. Vibration Isolation

If the controller is not properly designed, flexible bearing effects can lead to instability of the rotor nutation. That is, the nutational rotor may become unstable due to rotor energy losses.

Disk gyroscopics and flywheel rotor natural frequency effects create a plant which has characteristics that vary a great deal with operating speed. As the energy storage flywheel will vary over its speed range frequently, conventional feedback controllers will not be robust enough to stabilize the magnetic bearing system. A gain scheduled control algorithm is being developed for this application, similar to that developed for an earlier industrial flywheel design completed by UVa.

Imbalances in the wheel can create inertia forces which, when interacting with the stator, will transmit unwanted disturbances. Vibration isolation will be employed in this test rig controller, using synchronous open loop methods added to the feedback controller, to reduce the transmitted vibration to supporting structures and reduce the required bearing forces simultaneously. This technique has been employed for some time by UVa, however, it has not commonly been employed in industrially supplied magnetic bearing

units to date. A modification of the algorithm developed for industrial magnetic bearing supported compressors will be employed.

Challenge 5: Sensor and Controller Hardware

Inductive ring sensors, of the type developed by UVa previously for an industrial textile spindle, will be employed to determine the shaft position at suitable locations. Preamplifiers are required for the control hardware system. Existing UVa preamplifier designs will be adapted to this application. Commercial power amplifiers will drive the bearing coils. These are required to have the proper kVA requirements for the bearings and the necessary bandwidth. A software driven commercial packaging approach will be utilized to link the sensors and drivers through the controller to damp the dynamics of the flywheel rotor.

3.1.6.3 Results, Summary, Key Accomplishments, and Future Effort

GENERAL RESULTS

• Critical design parameters for the magnetic bearings for the EMFB include:

1. Small Size
2. Low Weight
3. Minimum Power Loss
4. Static Load Capacity
5. Dynamic Load Capacity
6. Shock and Acceleration Capability
7. Operating Speed Range
8. Hoop Stresses

These design parameters were specified by American Flywheel Systems documents.

• The rotor/magnetic bearing design has been completed. It includes:

1. Shaft
2. Magnetic Bearings
3. Sensors
4. Back-up Bearings

The magnetic bearing design has been integrated with the motor/generator and flywheel component designs. A full assembly drawing of the magnetic bearings/shaft and associated components has been developed. The rotor design includes the component assembly and disassembly procedure.

MAGNETIC BEARING DESIGN

• Both radial and thrust magnetic bearing designs were obtained. Five bearings were designed.

1. Electromagnetic Radial Bearings (2 bearings)
2. Gravity Load Thrust Bearing (1 bearing)
3. Electromagnetic Dynamic Thrust Bearing (2 bearings)

• Radial and thrust magnetic bearing designs have been developed. They have several important features that represent advantages over other, competing designs.

1. Design Meets Peak Load Capacity Requirements
2. Meets Operating Speed Requirements
3. Very Compact Design
4. Much Smaller Than Competing Designs
5. Allows for Overall Reduction In Flywheel Size
6. Very Low Weight Design

SENSORS

• Both radial and axial sensors have been designed for the magnetic bearing system.

1. Inductive Radial Ring Sensors
2. Inductive Axial Sensors

ROTOR DYNAMICS

• A rotor dynamics model of the shaft/magnetic bearing system has been developed. It is a beam model of the shaft with associated magnetic bearing components attached. The flywheel was modeled as a lumped mass in this initial design.

• Undamped critical speeds have been evaluated.

• Gyroscopic effects in the EMFB magnetic bearing shaft have been evaluated. They affect the magnetic bearing controller design.

• A reduced order model has been developed to simplify the rotor dynamics/control analysis.

CONTROLLER

• Several magnetic bearing controller designs have been evaluated. They are:

1. Proportional-Derivative
2. Tilt-Translate Control
3. PD/Notch Filter Control
4. Two Plane PD Control
5. μ Synthesis Control

• An advanced control method was employed to obtain the final controller design. It will control the rotor through the operating speed range. However, it has some limitations which could be significantly improved with additional development.

SUSPENSION DESIGN

• A complete rotor design incorporating the shaft, magnetic bearings, sensors, and back-up bearings was developed and presented in this report. It is integrated with the motor/generator and flywheel developed by other design teams and supplied by AFS. A full assembly drawing is presented in this section.

• The rotor design has been completed. The rotor/bearing design components can be assembled and disassembled (if necessary).

SUSPENSION DYNAMICS ANALYSIS

• A rotor dynamics analysis was carried out for the shaft/bearing unit. The shaft model included the magnetic bearing and motor/generator shaft with an equivalent disk model of the flywheel at the rotor center location.

• Undamped critical speeds were calculated for the rotor as a function of bearing stiffness. These must be considered relative to the expected EMFB operating range.

• Two rigid body modes were found by the analysis, where the shaft does not bend significantly. It is easily seen that the rotor passes through two of these rigid body modes at very low frequency, well below the minimum operating speed.

• It is desirable to change the current situation where two critical speeds are within the design speed range of the EMFB. There are several options for operation under this situation, if the vibration levels are large at these speeds: 1) these two critical speeds could be avoided during constant speed operation requiring the rotor to either accelerate or decelerate through them, 2) the magnetic bearing control system could be programmed to magnetically (open loop) balance the rotor at those speeds, or 3) the EMFB rotor could be modified to remove one or both of these critical speeds from the operating speed range. The most likely scenario is the combination of some of these options. For example, one of the critical speeds might be moved above the operating speed range and the other one open loop balance compensated by the magnetic bearing controller.

• Free-free mode shapes were evaluated for the rotor operating at maximum operating speed. Both forward and backward modes were plotted, including gyroscopic effects.

SUSPENSION SYSTEM DESIGN

A complete rotor (shaft) design was carried out for the shaft associated with the magnetic bearings, sensors, and back-up bearings. The design included the integration with the motor/ generator and flywheel. Figure 3.1.6-5 gives a drawing of the full flywheel design, with labels indicating the significant magnetic bearing, sensor and back-up bearing of the design. It may be noted that the flywheel is drawn in the horizontal position, simply to fit it on a page. However, the flywheel is designed to operate in the vertical position.

Figure 3.1.6-6 shows a close up view of the upper section of the magnetic bearing configuration. Note that the flywheel top is to the right of the drawing and the flywheel bottom is to the left. As noted earlier, the magnetic bearings, sensors, and back-up bearings can be disassembled, if necessary. The back-up bearing reception pad, sensor rotor, radial bearing rotor, and permanent magnet thrust disk are all permanently attached to the shaft. In the upper section, the back-up bearing and associated housing would be removed first. Then the sensor stator is removed. The magnetic radial bearing stator is removed next, sliding over the sensor rotor lamination stack. Finally, the permanent magnet thrust bearing stator is capable of being removed because it is larger in internal diameter than the shaft laminations. Finally, the shaft can be removed while still attached to the generator/motor shaft.

Figure 3.1.6-7 gives a zoomed in view of the lower section of the magnetic bearing configuration. It can be disassembled in similar fashion, if necessary. The axial proximity sensor and associated cap is removed first via the bolted arrangement shown. The back-up bearing housing, also containing the sensor stator, is removed next. The radial bearing stator and lower portion of the active thrust bearing slide out, over the sensor rotor component, radial magnetic bearing rotor component, and the backup bearing touch down pad. The active thrust bearing thrust disk can be loosened and removed with hydraulic pressure through the holes provided, as shown in Fig. 3.1.6-7. Finally, the upper stator portion of the active thrust bearing can be slid out. Following all of this disassembly, the shaft can be removed in either the upper or lower direction, if needed.

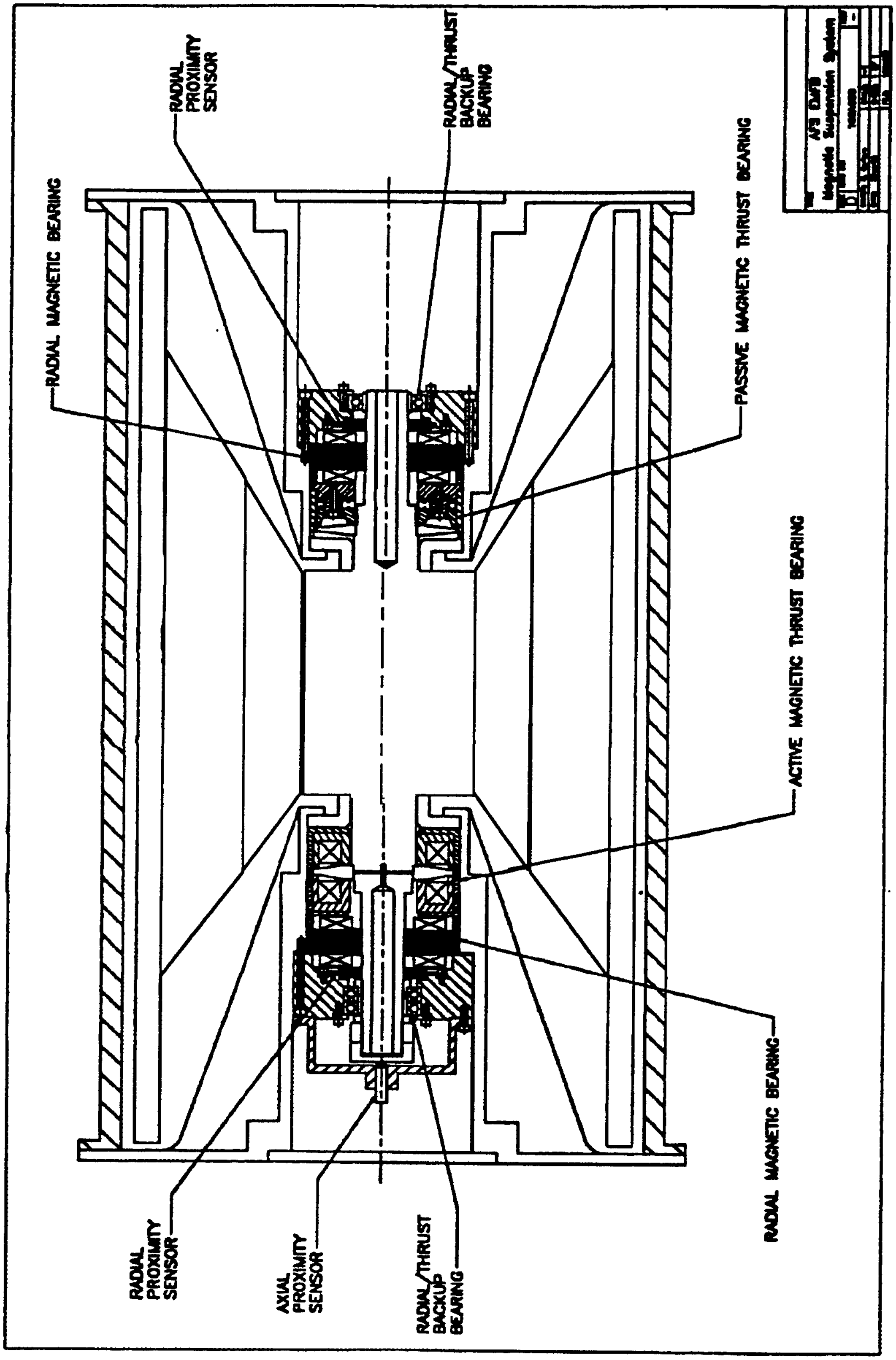


Figure 3.1.6-5 Typical Flywheel Assembly Drawing Showing Magnetic Bearing

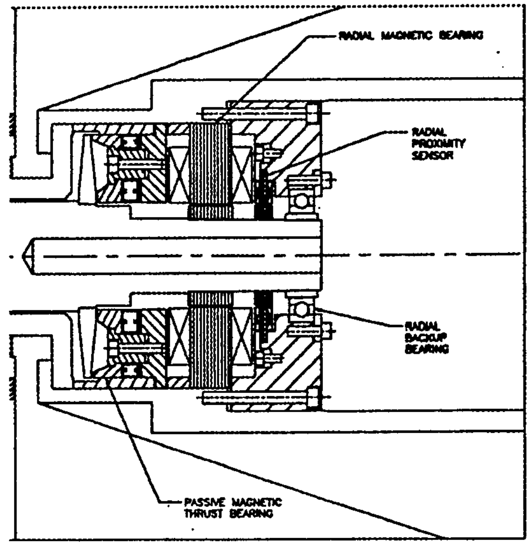


Figure 3.1.6-6 Cross-section of Upper Magnetic Bearing

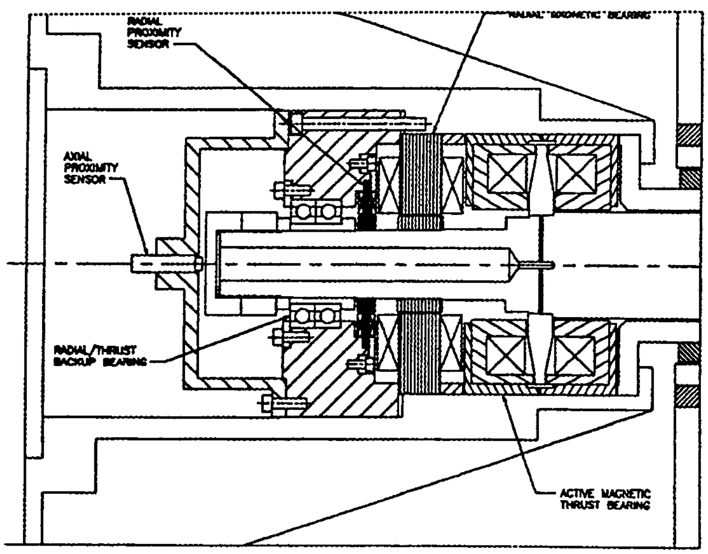


Figure 3.1.6-7 Cross-section of Lower Magnetic Bearing

3.2 References

References

2-1 Abacus Report for DOE dated April 30, 1993

3-1 AFS Report, Computer Feasibility Study into the use of an Advanced Energy Storage System, July 1990, ARMY BRADC, Ft. Belvoir, Va.

3-2 Honeywell Workplan (Proprietary), February 1993

3-3 SAE Report J1211